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INTERNAL-COMBUSTION ENGINES

THEORY AND DESIGN

BY

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SECOND EDITION

INTERNATIONAL STUDENT EDITION

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INTERNAL COMBUSTION ENGINES

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Preface to the Second Edition

During the twelve years that have elapsed since this book was written, the progress in the field of internal-combustion engines has continued, possibly even with an accelerated tempo. The design and construction of existing engines have been improved and changed; new and better designs have appeared on the market, with a very conspicuous trend toward higher speeds and greater power output.

Deeper and more correct understanding has been attained in respect to thermo- and hydrodynamical processes and also in respect to the strength and wear of various machine parts.

This is justification for a new, entirely revised edition of the book. The author has tried to improve the book for class use by taking into account not only the suggestions which he has received from instructors who have used his book in their classes but also his own teaching experience. As might be expected, the suggestions were of a greatly differing nature. Some wanted more material in respect to theory; others in respect to recent research; and several asked for strengthening the chapters devoted to design.

At first the author was afraid that complying with all these suggestions would make the book too bulky as a text for an undergraduate course. However, since the book, even in the first edition, contained slightly more material than could be covered satisfactorily in a one-semester course, the author decided to include all material which seemed to be of interest from various points of view but to present it in such a way that certain chapters could be omitted without harm to the fundamental exposition. This gives a valuable flexibility to the book as a class text: it may be used in undergraduate courses, whether short or long, by covering only those chapters which the individual instructor considers most essential; the rest of the chapters may be used in a second, advanced course. The book may also be used as a text in a course consisting of classes in theory and in a design laboratory. Finally, the book has enough material for two consecutive courses, the first of a theoretical nature, followed by a second course dealing with the design of internal-combustion engines, the latter course to be in conjunction with a corresponding laboratory.

In spite of the increase of the amount of material included in this revised book, the author hopes that he has succeeded in his attempt to follow the general tendency toward fundamentals rather than the tendency toward specialization in engineering education. The theoretical discussion has been strengthened, and the chapters dealing with the design of parts are presented as a basis for similar design and not just filled with practical and empirical data. The order of presentation was rearranged with the purpose of further improving the teachability of the text. The problems have been revised and increased in number from 225 to 543.

The author wishes to acknowledge again the assistance given him by engine builders who supplied drawings and data pertaining to their engines. He also acknowledges the assistance received from instructors who have used the first edition of the book in their classes and who have sent in suggestions for its improvement.

V. L. MALEEV.

HOLLYWOOD, CALIFORNIA,
July, 1945.

Preface to the First Edition

THE beginning of this book dates back a quarter of a century when the author started his course in gas- and oil-engine design at the Institute of Technology of Tomsk (Russia). After being engaged in teaching and research work for a number of years, the author came to America, where circumstances made necessary a change to industrial engineering. For over ten years he worked as a designing, testing, and experimental engineer and finally as head of the engineering department of a concern building gas and oil engines.

During his work as a practical engineer he often felt the need for a book containing all the necessary data pertaining to the design and performance of oil and gas engines. But he felt even oftener the lack of a book giving a general foundation for the proper understanding of the principles involved in the design and operation of internal-combustion engines. The existing books, he found, were unsatisfactory for the purpose because they dealt with design alone or were too theoretical or of a purely descriptive nature.

When three years ago the author returned to teaching, he decided to complete this book as soon as possible, combining in it his experiences as both college instructor and practical engineer.

The field of internal-combustion engines is so broad and expands so rapidly that it cannot be covered in a single volume. However, the fundamentals of internal-combustion engineering can be given in one book so that the reader may become familiar not only with *what* has been done and *how* it is done, but also with *why* it should be done in a certain way.

The author hopes that his book will fill this purpose and thus will be equally useful to college students and to engineers and designers in the industry.

The plan of the book assumes that the reader is familiar with thermodynamics and engineering mechanics. However, to make the book a self-contained unit, a review is included of those parts of thermodynamics which are essential for a proper understanding of the theory of internal-combustion engines. With the same object in view most formulas based on thermodynamics and mechanics are not simply stated in a final form but derived from basic, more familiar equations.

The author wishes to acknowledge the assistance given him by engine builders who supplied drawings and data pertaining to their engines, and to mention particularly the Western-Enterprise Engine Company whose diversified lines of internal-combustion engines—gas, oil, and gasoline, stationary, marine, and airplane—allowed the selection of a great number of examples. The name of the manufacturer has not been mentioned every time a drawing was used. References to technical literature are given in the text and in the Appendix.

Finally the author acknowledges the assistance received from Professors H. M. Jacklin, C. M. Leonard, and V. W. Young in the form of constructive criticism of the manuscript.

V. L. MALEEV.

HOLLYWOOD, CALIFORNIA,
June, 1933.

List of Abbreviations

abs	absolute
°API	degree(s) American Petroleum Institute, specific gravity
atm	atmosphere
°Bé	degree(s) Baumé, specific gravity
bhp	brake horsepower
Btu	British thermal unit(s)
cc	cubic centimeter
cfm	cubic feet per minute
cu	cubic (in. or ft)
F	degree(s) Fahrenheit
fpm	feet per minute
fps	feet per second
fpsps	feet per second per second
ft	foot, feet
ft-lb	foot-pound(s), work
gpm	gallon(s) per minute
hp	horsepower
hr	hour(s)
ihp	indicated horsepower
in.	inch(es)
lb	pound(s)
lb-ft	pound-foot, moment of force or couple
mep	mean effective pressure
min	minute(s)
mip	mean indicated pressure
mph	mile(s) per hour
psf	pound(s) per square foot
psi	pound(s) per square inch
psia	pounds per square inch absolute
psig	pounds per square inch gage
R	degree(s) Rankine, Fahrenheit absolute
rpm	revolution(s) per minute
sec	second(s)
sp gr	specific gravity
sq	square (in. or ft)
vib	vibration

List of Symbols

a	acceleration, linear, fpsps
a, A	area, sq in. or sq ft
d, D	diameter, in.
c	specific heat, Btu/lb-°F
c_p	specific heat at constant pressure, Btu/lb-°F
c_v	specific heat at constant volume, Btu/lb-°F
c, C	a constant
C	chemical energy, Btu/lb
e	excess air, lb/lb
e_{sc}	excess scavenge air, lb/lb
E	modulus of elasticity, direct, psi
f	coefficient of friction
F	force, lb
g	acceleration of force of gravity, 32.2 fpsps
G	modulus of elasticity, shear, psi
h	enthalpy, Btu/lb
h	film coefficient, heat transfer, Btu/sq ft-°F-hr
k	ratio of specific heats, c_p/c_v
k	thermal conductivity. Btu/sq ft-°F-hr
l, L	length, also stroke, in.
m	mass. lb
M	molecular weight
M	moment of a force or couple, lb-in. or lb-ft
n	exponent of a polytropic curve
n	revolutions per minute
N	horsepower
N_b	brake horsepower
N_i	indicated horsepower
p	pressure, psi or psf
p_e	mean effective pressure, brake, psi
p_i	mean indicated pressure, psi
q	heat, Btu/lb
Q	total heat, Btu or Btu/hr
Q	heat value, Btu/lb or Btu/cu ft
Q_h	higher heat value, Btu/lb or Btu/cu ft
Q_l	lower heat value, Btu/lb or Btu/cu ft
r	compression ratio, volumetric
R	gas constant, ft-lb/lb-°F
s	entropy, Btu/lb-°F
S	total entropy, Btu/°F
s	stress, tensile or compressive, psi
s_s	shear stress, psi
S_e	elastic limit, psi
t	degrees Fahrenheit, °F
T	degrees absolute, Rankine, °R
T	torque, lb-in. or lb-ft
u	internal energy, Btu/lb
U	total internal energy, Btu
v	volume of 1 lb—specific volume, cu ft/lb
V	total volume, cu ft
v	velocity, fps
w	specific weight, lb/cu ft
W	weight, lb
W_k	work, ft-lb
α	contraction coefficient
$\alpha,, \beta, \gamma$	angles
γ	specific gravity, compared to water
η	efficiency
η_{ch}	charge efficiency
η_{id}	ideal efficiency
η_m	mechanical efficiency
η_{sc}	scavenge efficiency
η_v	volumetric efficiency
μ	discharge coefficient
ς	density, mass per unit of volume, lb/cu in.-in. psps
τ	time, sec, min, or hr
φ	velocity coefficient
ω	angular velocity, radians per sec

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INTERNAL-COMBUSTION ENGINES

CHAPTER 1

INTRODUCTION

1-1. General.—Internal-combustion engines use the same source of energy as steam engines and steam turbines. There are many reasons why internal-combustion engines are not only competing with other heat engines but in many instances are dominating the field. The class of service is the principal factor in several cases. The most outstanding application of internal-combustion engines is for transportation—in automobiles, trucks, and airplanes. In all these cases simplicity of operation and light weight are the deciding factors. In many small and medium-size stationary power plants, on farms and in small industrial enterprises, simplicity and low cost of operation are governing the adoption of gas and oil engines in preference to steam engines or electrical motors. In larger power plants, used for production of electric current or propulsion of boats and ships, and in locomotives, the fuel economy decides the choice in favor of oil engines.

A better utilization of the heat energy of fuel is an important feature of internal-combustion engines. The expansion in the use of internal-combustion engines is also materially assisted by their adaptability to different fuels—gaseous, liquid, and solid.

As an introduction to problems encountered in the selection, operation, and design of internal-combustion engines, a classification and brief discussion of representative engines are given first.

The principles of thermodynamics and their application to the theory of internal-combustion engines are reviewed next.

Conditions and means of economical utilization of the heat of combustion are considered afterward in connection with the peculiarities of different fuels. Problems of combustion in actual engines are analyzed next. After that, methods of determination of the main engine dimensions and guiding points in the designing of the principal engine parts are brought out.

The closing chapters discuss the main points connected with rating and testing of internal-combustion engines and the principal features of the new form—the gas turbine.

1-2. Classification of Engines.—Internal-combustion engines can be divided into several groups using different features as characteristics: operating cycle, method of charging the cylinder, fuel used, general design; the latter in turn includes method of ignition, position and number of cylinders, rotative speed, etc. These classifications often overlap, since engines in the same class from one point of view belong to different classes from another point of view.

Method of Charging.—In regard to the method of charging the cylinder, all engines can be divided into two groups, *four-stroke* and *two-stroke* engines, depending upon the number of strokes required for completion of one cycle. Another designation, four-cycle and two-cycle engines, although often used, is incorrect and meaningless.

Fuel.—In regard to fuel, engines may be divided into three classes, according to whether they operate on gaseous fuel, liquid fuel, or solid fuel. Liquid-fuel engines may be subdivided into those using volatile fuels, such as gasoline or alcohol, and those using heavy oil; the first vaporize the fuel by a carburetor, and their method of operation is very similar to that of gas engines.

In heavy-oil engines the fuel is injected into the air charge toward the end of the compression stroke. These engines may be divided into low-, moderate-, and high-compression engines. In low-compression engines the ignition is obtained by means of a hot surface, such as a hot bulb, or an electric spark. Moderate-compression engines run with the ignition produced by the compression but require a special igniter for starting. In high-compression engines the ignition is effected by the compression alone; no outside means are required for starting.

The development of mechanical-injection systems showed that the compression-ignition engines are not more expensive, nor do they require more skilled operators than the moderate-compression engines. As a result, practically all builders of moderate-compression engines and some builders of low-compression engines changed their designs and raised the compression to obtain ignition by compression under all conditions. The present low-compression engines are chiefly of the cheap two-stroke type often called *semi-diesel engines*, although this name does not mean anything and is used only as a sales appeal.

Engines operating on solid fuel, mostly coal, so far are using it after it has been gasified in gas producers; thus they are simply gas engines. Various forms of gas producers have been developed to make solid fuels available for use. Almost forty years ago Rudolph Diesel tried unsuccessfully to use pulverized coal, injecting it directly into the engine cylinder. However, a German designer, R. Pawlikowsky, has developed an engine, similar to Diesel's original one, that operates on pulverized coal. The main handicap to the commercial success of this engine is the excessive wear of the cylinder liner, piston, and piston rings, due to the abrasive action of the particles of ash formed during combustion. When and if this wear is reduced to a practical limit, the engine will have a great future because the supply of coal is incomparably greater than the supply of liquid fuels.

Ignition.—At present the majority of engines operate with electric ignition and therefore are designated as *spark-ignition* engines. High-compression oil engines do not need any special ignition mechanism, the fuel being ignited when it comes in contact with the highly compressed and therefore hot air charge; hence their designation as *compression-ignition* engines.

General Design.—In regard to the general design, engines may be classed as single- and double-acting. The latter design is used only for large engines. Other classifications under this head are horizontal- and vertical-type engines, meaning engines with horizontal or vertical center lines of the cylinders, also single- and multicylinder engines, with two, three, four, and more cylinders.

In *V*-type engines the cylinder axes are located in two intersecting planes, which permits the pistons of each pair of cylinders to act upon the same crank. This arrangement shortens considerably the length of the crankshaft and of the whole engine. In *radial* engines this scheme is carried to its logical limit, the crankshaft having only one crank, upon which act five, seven, or nine pistons.

1-3. Four-stroke Spark-ignition Engines.—Such an engine and its cycle of operation are shown diagrammatically in Fig. 1-1*a*–1*d*.

During the first, or compression, stroke, Fig. 1-1*a*, the piston moves toward the cylinder head, compressing the charge and thus preparing it for ignition, which occurs through an electric spark from the spark plug *sp* near the end of the stroke. The ignition is followed by an almost instantaneous combustion, accompanied by a considerable pressure rise. During the second, working, or expansion, stroke, Fig. 1-1*b*, the piston is pushed by the gases toward the opposite cylinder end; slightly before dead center the exhaust valve *e* begins to open, releasing the burned gases to the atmosphere.

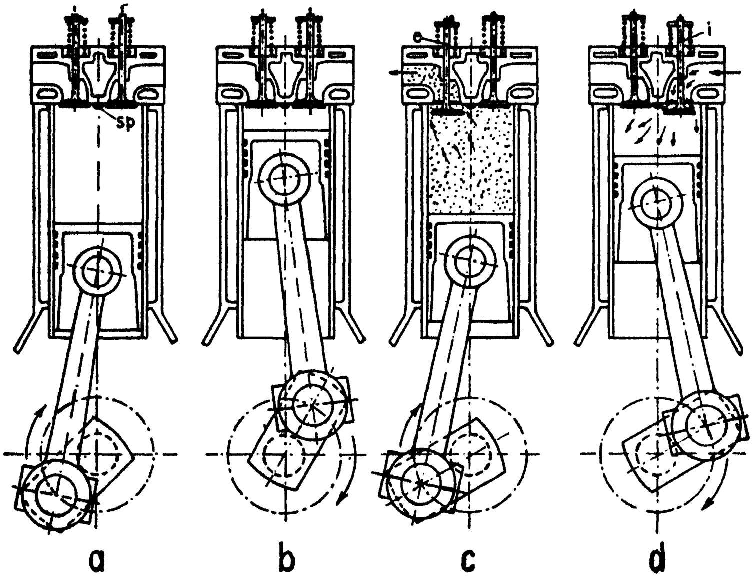

FIG. 1-1.—Events in a four-stroke cycle.

During the third, or exhaust, stroke, the piston moves back toward the cylinder head and pushes the burned gases out, Fig. 1-1*c*, through the valve *e*. At the beginning of the fourth, intake, or suction, stroke, Fig. 1-1*d*, the exhaust valve *e* is closed and the intake valve *i* is opened. During this stroke the piston moves away from the cylinder head, creating a vacuum which causes the cylinder to be filled with a fresh charge consisting of a mixture of air and fuel, gas or gasoline vapor, through the valve *i*.

Figure 1-2 shows a typical indicator diagram of a spark-ignition four-stroke engine: 1-2 corresponds to the compression stroke; 2-3 is combustion at nearly constant volume, often termed *explosion;* 3-4 is the expansion on the power stroke; 4-5 is the exhaust; and 5-1 is the suction stroke.

Figure 1-3 shows a modern six-cylinder gasoline-truck engine with the names of all main engine parts.

Figures 1-4 and 1-5 show a typical modern passenger-car engine. It has six L-head cylinders with a 3½-in. bore and 4¼-in. stroke; the crankshaft is of rigid construction and is supported on four main bearings having a 2¾-in. diameter. The connecting rods are drilled for pressure lubrication of the full-floating wristpins. The pistons are of aluminum with split skirts. Instead of a section through the sixth cylinder, Fig. 1-5 gives a section through the valves, showing the spark plug, the exhaust valve under it, and the slightly larger intake valve. However, in many automobile engines the intake and exhaust valves are made of the same size. Figure 1-4 shows the crankshaft and one of the valves with its mechanism, the shape of the combustion space, and two sections of the pressure-oil gear pump. The engine has a compression ratio of 6.71:1 and develops 73 and 105 hp at 2000 and 3600 rpm, respectively. These speeds correspond to a car speed of 37.2 and 66.9 mph.

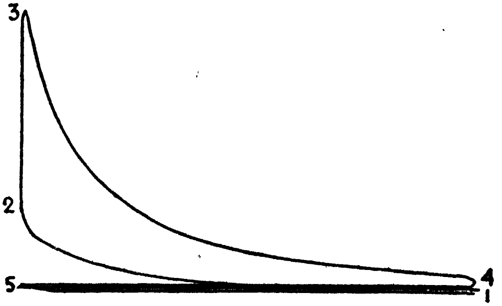

FIG. 1-2.—Four-stroke gas engine diagram.

Figure 1-6 shows a typical double-acting tandem gas engine arranged for direct coupling to a compressor or blower. When this engine is used for driving an electric generator or other rotary machinery, the tie rods *a,a* and the auxiliary crosshead *b* are omitted and the bedplate is shortened, as shown by dotted lines *c*.

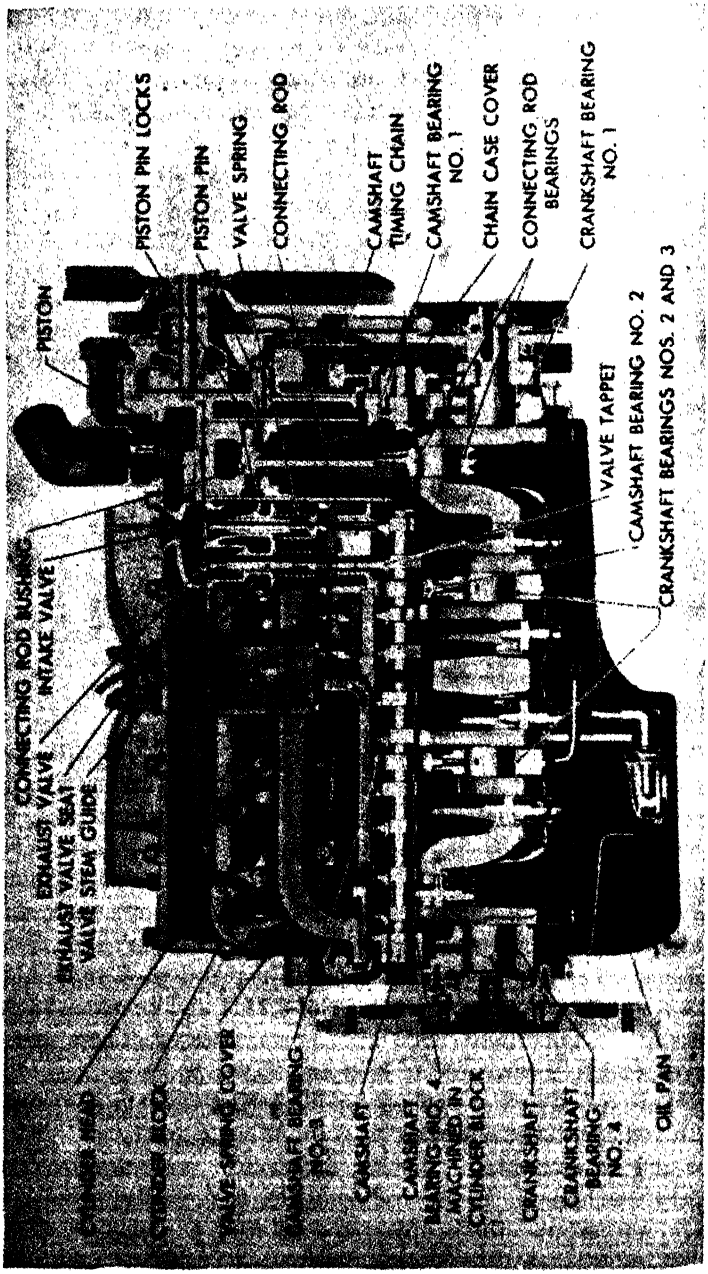

Fig. 1-3.—Six-cylinder automobile engine with engine-part nomenclature.

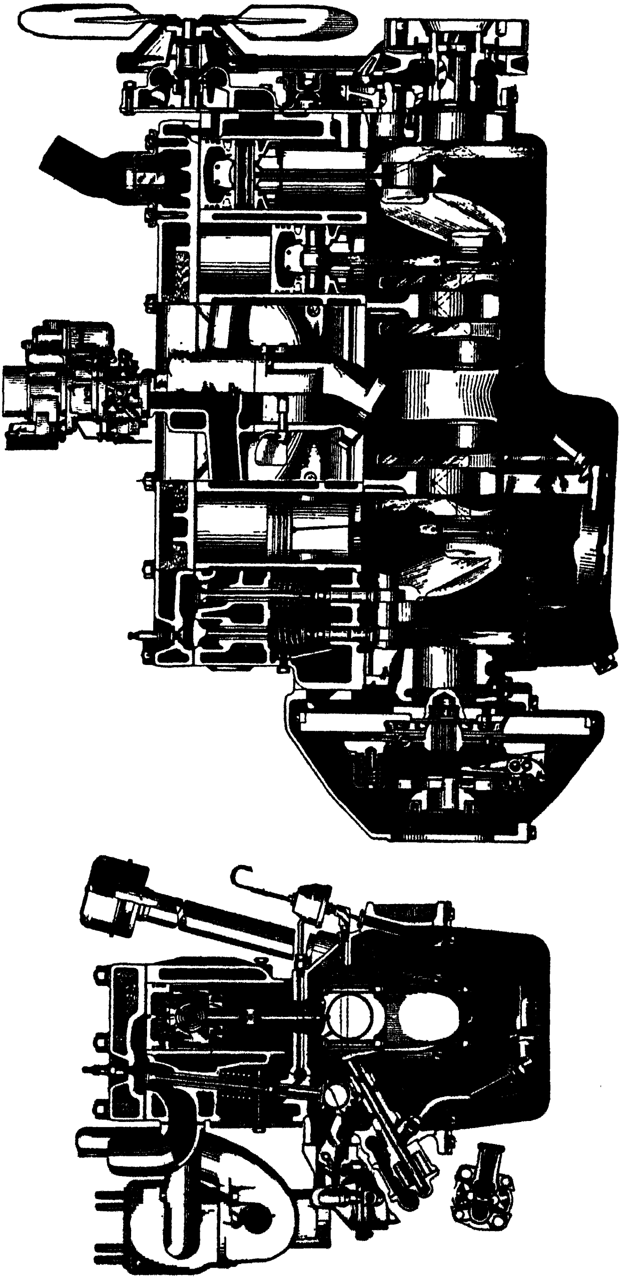

Fig. 1-4. and 1-5.—Cross section and longitudinal section of 1942 Packard passenger-car engine. (*Courtesy of Packard Motor Car Company.*)

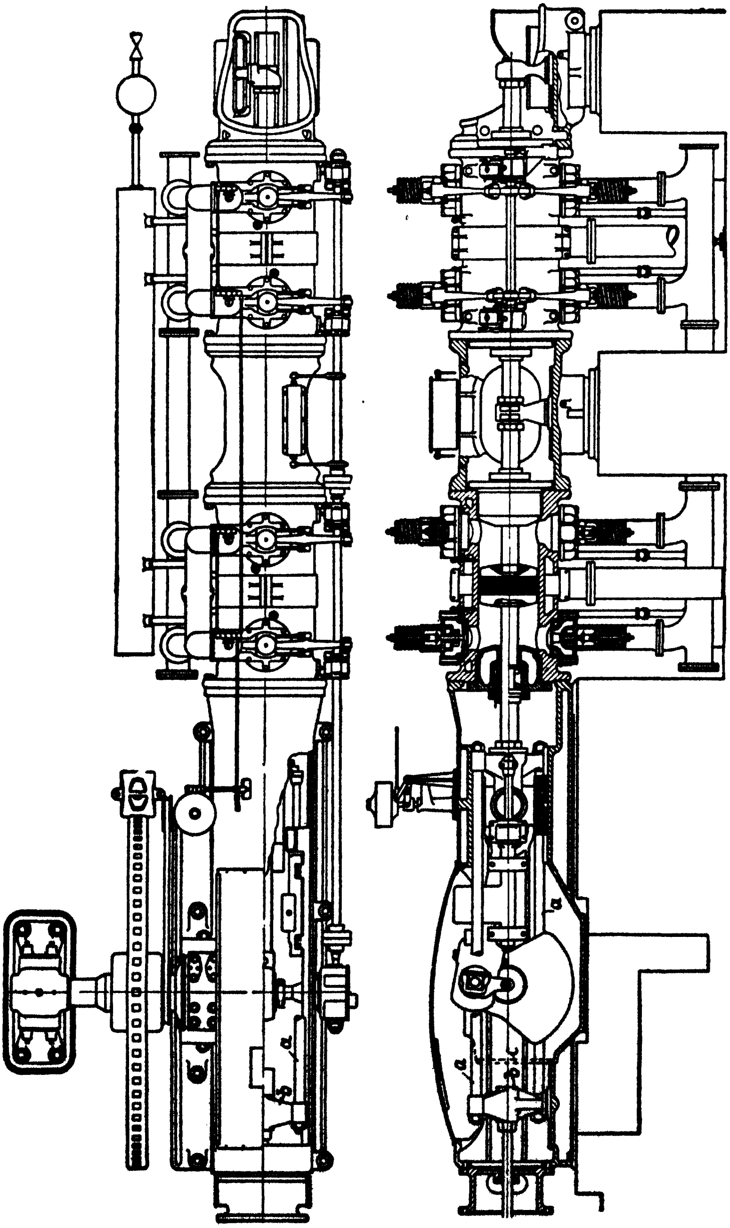

FIG. 1-6.—Worthington 330-hp double-acting tandem gas engine.

The largest gas engines built in this country, which develop 10,000 bhp, are made as double-tandem engines, each side resembling the general design of Fig. 1-6.

1-4. Two-stroke Spark-ignition Engines.—In such an engine the cycle is completed in two strokes, or one revolution of the crankshaft, as against two revolutions in a four-stroke-cycle engine. Thus, all other factors being equal, a two-stroke engine will develop, at least theoretically, twice the horsepower of a four-stroke engine having the same piston displacement.

The difference between two-stroke and four-stroke engines is in the method of removing the burned gases and filling the cylinder with a fresh charge. In a four-stroke engine these operations are performed by the engine piston during the exhaust and suction strokes. In a two-stroke engine these operations are performed near the dead center, after the expansion stroke, by a stream of air or air-fuel mixture admitted under pressure and delivered by a separate pump.

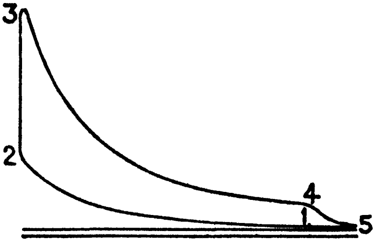

FIG. 1-7.—Two-stroke gas-engine diagram.

Figure 1-7 represents a typical two-stroke-engine indicator diagram. Compression line 1-2, combustion 2-3, and expansion 3-4 do not differ from the corresponding parts of a four-stroke diagram. At point 4 the engine piston begins to uncover special exhaust ports, and the bulk of the burned gases escapes to the atmosphere. After the pressure has dropped almost to atmospheric, air, slightly compressed by a so-called *scavenge pump*, is admitted into the cylinder, scavenges the latter, and expels most of the remaining burned gases. The scavenging usually lasts until point 1, when the piston, moving toward the cylinder head, covers the scavenge and exhaust ports and compression starts. In gas engines a separate pump delivers the gas into the cylinder either slightly before point 1 or, in the newest constructions, after point 1, thus avoiding any escape of gas into the exhaust.

Figure 1-8 shows diagrammatically a two-stroke spark-ignition engine: Toward the end of the expansion stroke the piston first uncovers the exhaust ports *e*, the gases begin to flow out, and the pressure drops; soon after that the piston uncovers the so-called *scavenge ports s*, and the lightly compressed air-fuel mixture is admitted to the cylinder, expelling the remaining burned gases. On the return stroke the piston covers first the scavenge ports *s*, then the exhaust ports *e*, and the compression starts. Near the upper dead center the charge is ignited by a spark from the spark plug *sp*.

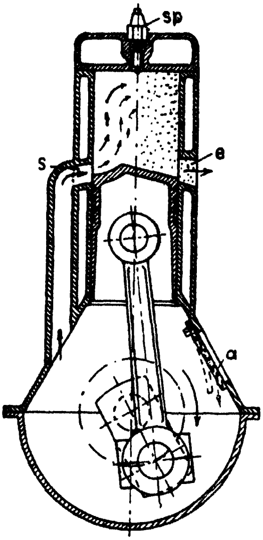

Fig. 1-8.—Crankcase scavenge spark-ignition engine.

Figure 1-9 shows a horizontal two-stroke gas engine of the so-called *crankcase* scavenging type. The scavenging air enters the crankcase through a check valve *v* while the piston is moving to the left, and on the return stroke is compressed until the scavenge ports *s* are uncovered by the piston. The flow of scavenge air continues until on the return stroke the piston covers the scavenge ports. The exhaust gases escape through the exhaust ports *e*. This scheme has the advantage of simplicity and is used extensively. However, the volumetric efficiency of the crankcase pump is rather low, and not enough air is

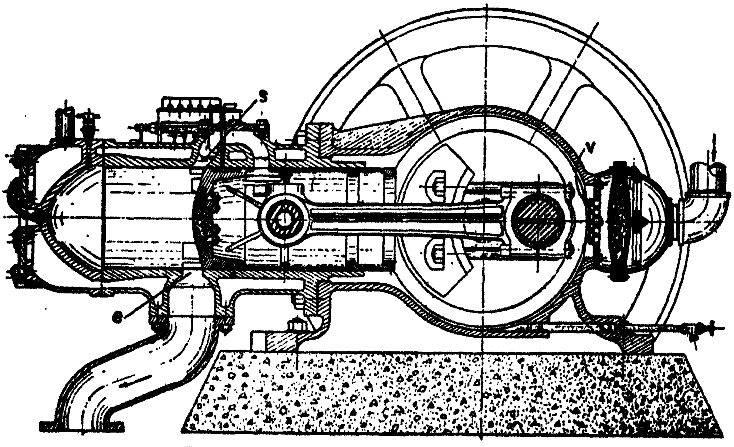

Fig. 1-9.—Weber two-stroke gas engine.

furnished for a good scavenging. Owing to the limited amount of air admitted to the cylinder, a correspondingly small amount of fuel can be burned and the engine, in proportion to its piston displacement, develops a relatively small power. By using a separate pump to furnish the scavenging air the engine output can be increased.

1-5. Compression-ignition Engines.—The first commercially successful compression-ignition engine was developed by Diesel. The modern compression-ignition engines differ in so many respects from the original diesel engine that it is more correct to divorce them from the name of Diesel and simply to call them *compression-ignition engines.* They operate on either the four-stroke or the two-stroke cycle.

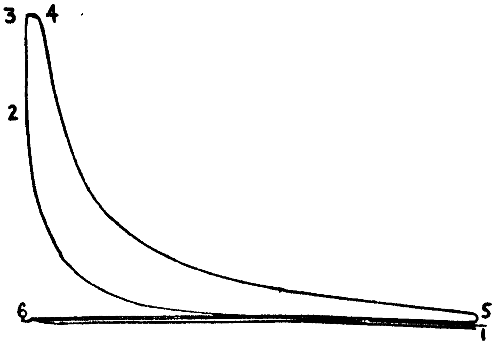

FIG. 1-10.—Diagram of a four-stroke compression-ignition oil engine.

The main differences between a spark-ignition and a compression-ignition-engine cycle are as follows: In the former the combustible mixture of fuel and air is introduced into the engine cylinder either before the compression stroke begins or shortly after. A low compression ratio must be used, to prevent the temperature at the end of the compression from rising to the ignition temperature of the charge; also, in this type, ignition is started by an electrical spark. In a compression-ignition engine, on the other hand, only air (with a small amount of residue gases) is compressed, and the fuel, as a finely atomized liquid, is injected shortly before the piston reaches the dead center. The temperature of the compressed air is higher than the ignition temperature of the fuel, and the latter begins to burn without any outside means of igniting it.

An indicator diagram, Fig. 1-10, explains the operation of a mechanical-injection four-stroke compression-ignition engine. During the compression stroke 1-2, the pressure and with it the temperature of the air within the engine cylinder are raised sufficiently high to ignite the fuel oil when it is injected slightly before the dead center. The burning of the fuel at first raises the pressure, line 2-3, and then for a short period maintains an approximately constant pressure, line 3-4, while the piston begins the power stroke. The expansion line 4-5, exhaust 5-6, and intake 6-1 do not differ from the corresponding events in a spark-ignition engine.

In a two-stroke compression-ignition engine, the compression and power strokes are identical with those of a four-stroke engine, Fig. 1-10, and the scavenging procedure does not differ from that of a spark-ignition engine, as illustrated by Fig. 1-7.

A few drawings with a corresponding description will explain more in detail the operation of these engines.

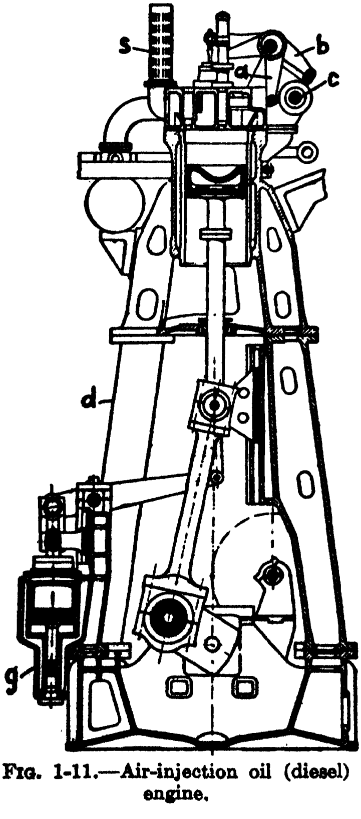

FIG. 1-11.—Air-injection oil (diesel) engine.

Figure 1-11 gives a cross section of a low-speed air-injection diesel engine as used in large motor ships. The intake and exhaust valves are operated by the camshaft *c*, with cams actuating the rocker arms *a* of the fuel valve and *b* of the intake valve. Air is admitted through silencers *s*, and the exhaust gases are discharged into the exhaust manifold *e*; *g* is the three-stage air compressor.

Figure 1-12 shows a medium-speed mechanical-injection oil engine of the so-called *constant-pressure* type. The top of the trunk piston *p* is so turned as to form a ring-shaped open combustion chamber. The fuel is pumped into the fuel-piping system, from which it is admitted to the fuel valve *f*. The latter is lifted by a rocker-arm mechanism actuated by means of a cam and push rod. The engine is started by compressed air admitted to the cylinder through the starting valve *m*, which is

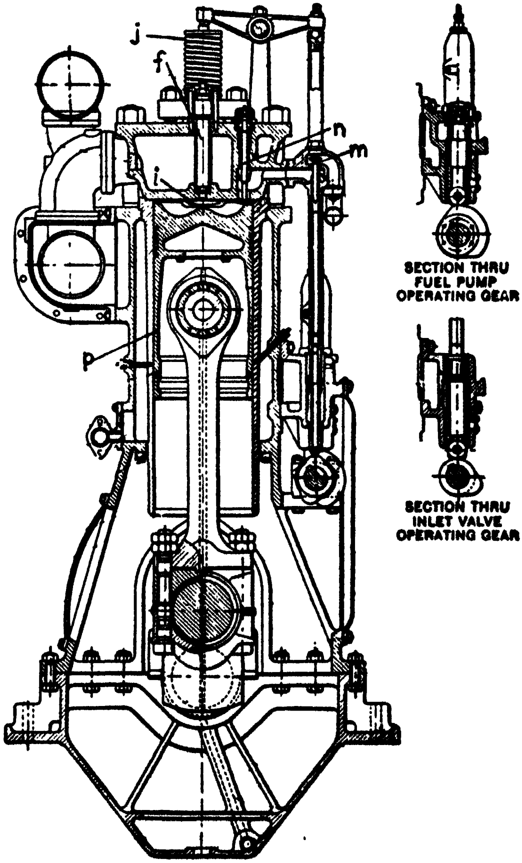

Fig. 1-12.—Mechanical-injection oil engine. *Courtesy of Worthington Manufacturing Company.*

actuated mechanically, and the check valve *n* in the cylinder head. To clear the inlet valve *i* and the symmetrically located exhaust valve not shown in the drawing, the top of the piston is recessed; *j* is the spring which closes the inlet valve.

Figure 1-13 shows another medium-speed mechanical-injection oil engine with a so-called *turbulence head*. During the com-

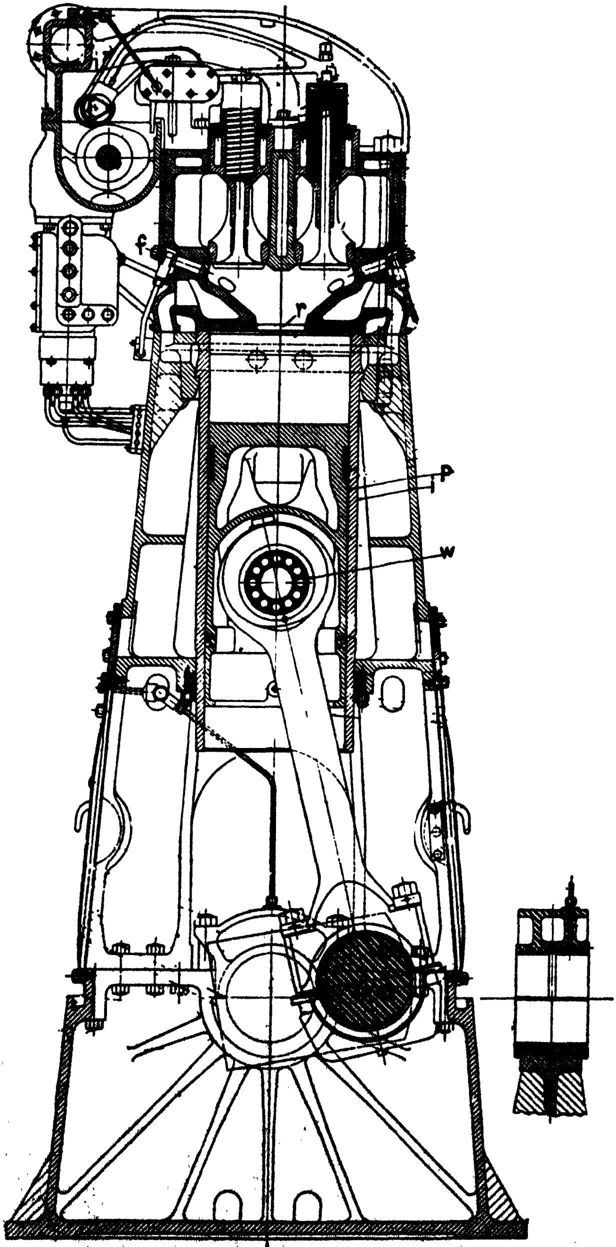

FIG. 1-13.—Price-Rathbun airless injection oil engine.

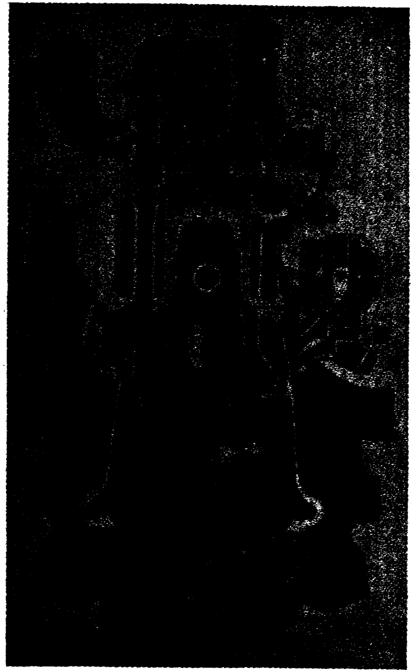

FIG. 1-14.—Cross section of a compression-ignition oil truck engine.

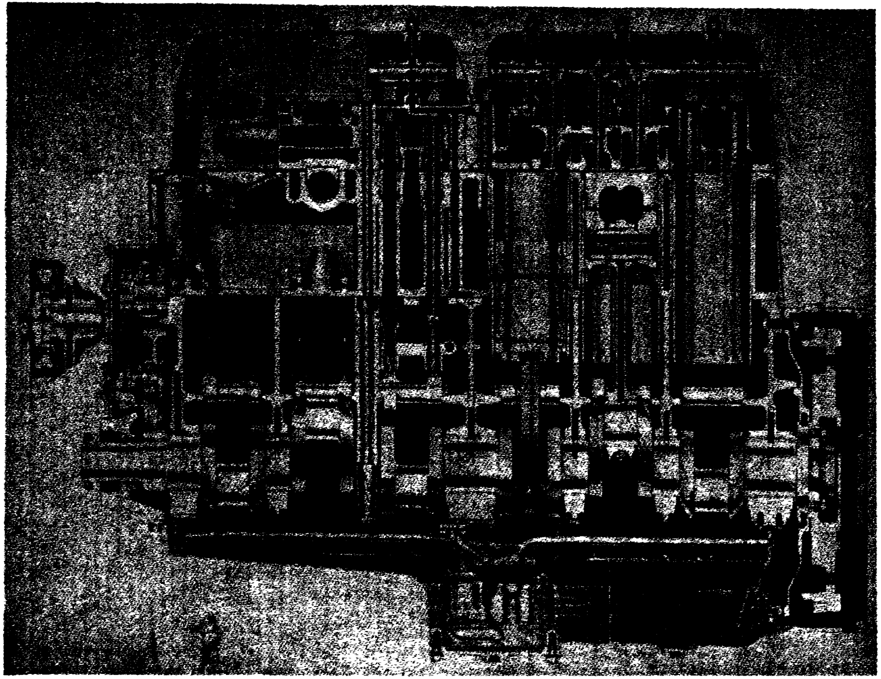

FIG. 1-15.—Longitudinal section and part view of a compression-ignition oil truck engine. (*Courtesy of Hercules Motors Corporation.*)

pression stroke the flat-top piston pushes the air through a restriction *r* into the combustion chamber, creating a turbulence that assists the burning of the fuel. The two fuel nozzles *f*, *f* also help to distribute the fuel through the combustion space. In this construction the center lines of the intake and exhaust valves are located in a plane normal to the crankshaft center line and therefore are visible on the cross-sectional drawing.

Figures 1-14 and 1-15 show a compression-ignition truck engine with a turbulence chamber. The fuel is injected by the pump *p* through the fuel injector *f* into the spherical chamber *c*, in which the piston during the compression stroke produces a strong turbulence; *g* is a glow plug to facilitate starting in cold weather.

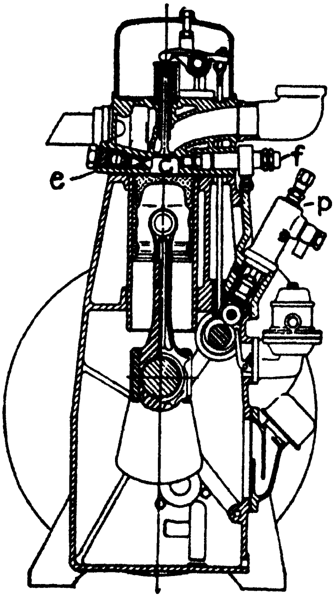

FIG. 1-16.—Oil engine with an energy cell.

Finally, Fig. 1-16 shows a small high-speed compression-ignition oil engine with a so-called *energy cell.* The fuel pump *p* delivers the fuel to the fuel nozzle *f*, which injects it into the combustion space *c*, where part of the fuel starts to burn; the core of the spray, however, enters the energy cell *e*, which consists of a pear-shaped space connected through a restriction with a spherical part. Some of the fuel which enters the energy cell ignites there, and the ensuing rapid combustion produces a stream of gases back into the main combustion chamber, creating a turbulence there and thus accelerating the combustion.

Some of the two-stroke compression-ignition oil engines are built along the same lines as two-stroke gas engines. In fact the engine shown in Fig. 1-9 is a convertible engine: by putting on a new cylinder head, which gives a higher compression ratio, removing the spark plug and magneto and inserting a fuel nozzle, and by attaching a fuel pump, the gas engine is converted into a compression-ignition oil engine.

Figure 1-17 shows the upper part of a modern two-stroke high-speed compression-ignition oil engine. The exhaust is released and pushed out through two symmetrically located poppet valves in the cylinder head; the air for scavenging is furnished by a built-in three-lobe rotary blower and is admitted through two rows of round-hole ports controlled by the piston edge. The fuel injector is located in the center of the cylinder head between the two exhaust valves.

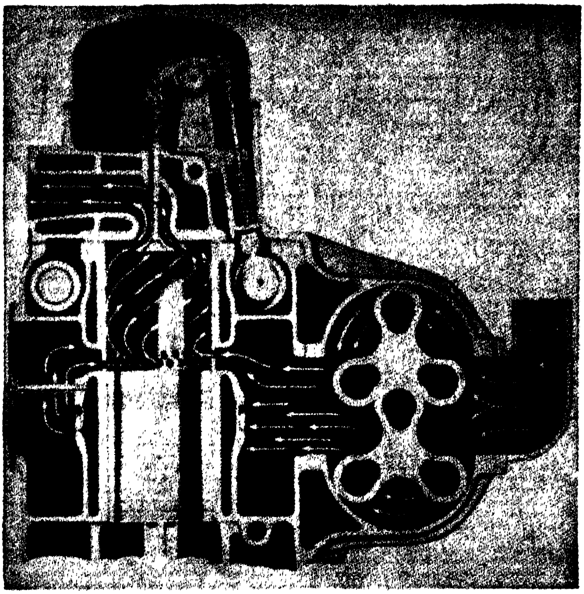

FIG. 1-17.—Uniflow scavenge system of a General Motors compression-ignition oil engine.

Figure 1-18 shows a large medium-speed two-stroke-cycle engine in which both the exhaust and the scavenging are accomplished through ports uncovered by the piston edge. In very large units the scavenge air is furnished by a separate small oil engine.

Figure 1-19 shows another large medium-speed engine with a different type of scavenging scheme. The engine is used in many stationary power plants and in motor ships.

Finally, Fig. 1-20 shows a large double-acting mechanical-injection engine built primarily for marine use, but also used in stationary power plants and as stand-by for peak loads in steam-turbine power plants.

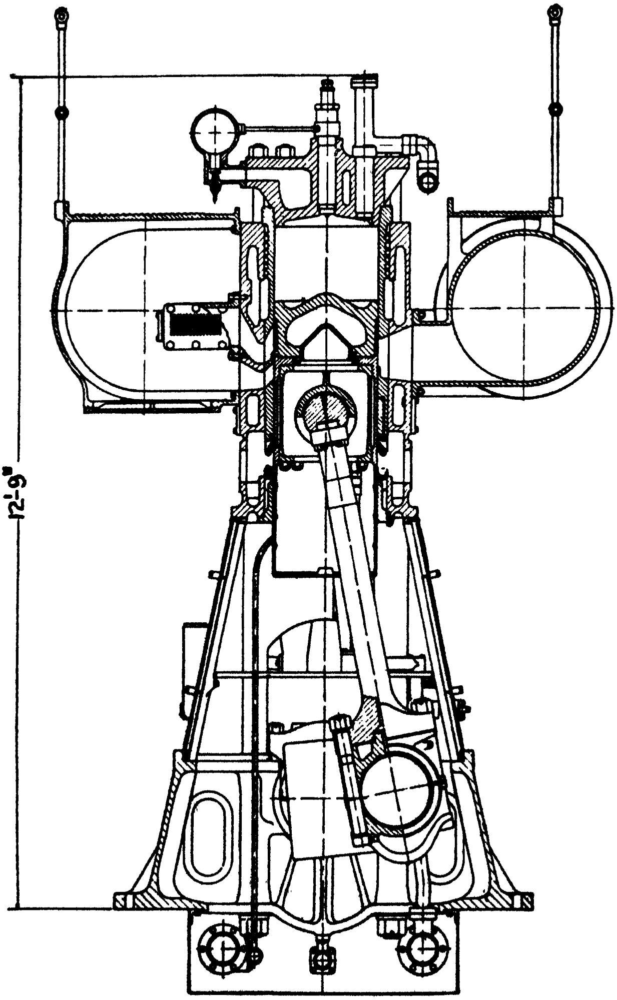

FIG. 1-18.—**Busch-Sulzer two-stroke oil engine, 300 hp per cylinder.** *(Courtesy of Busch-Sulzer Bros. Diesel Engine Co.)*

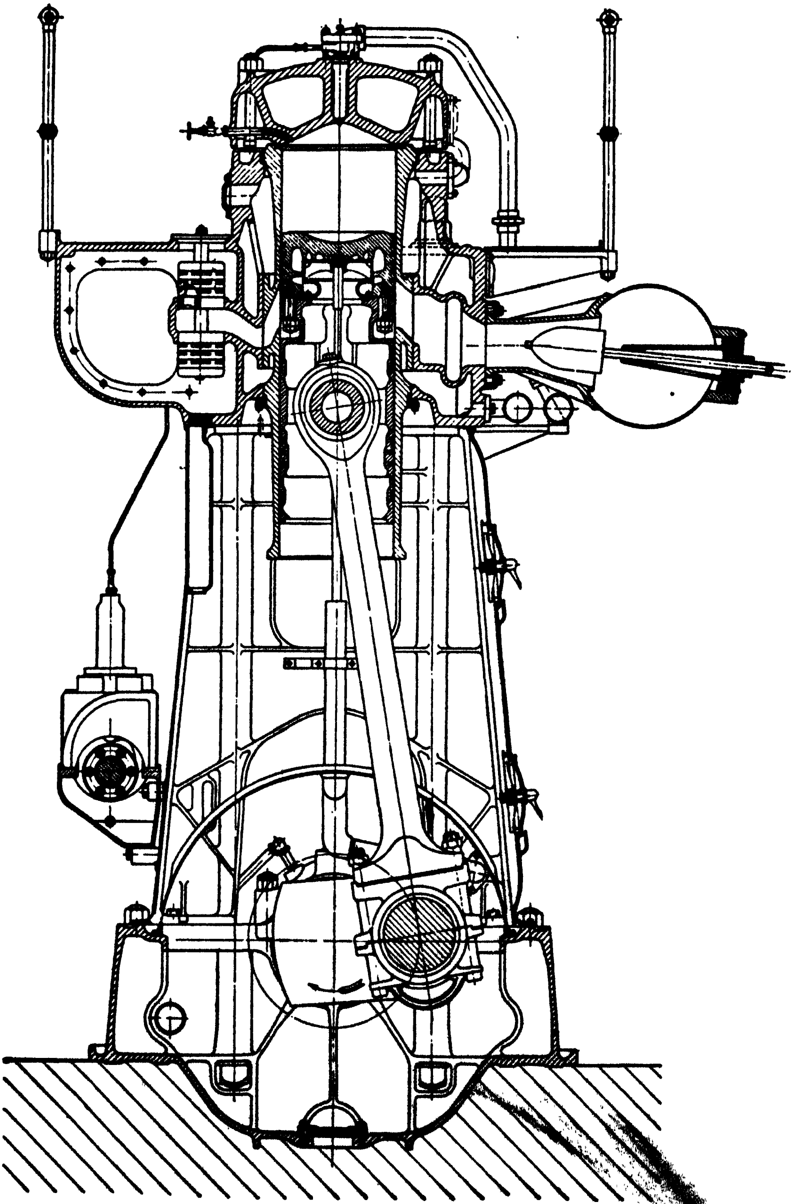

Fig. 1-19.—Nordberg two-stroke oil engine, 425 hp per cylinder. (*Courtesy of Nordberg Manufacturing Company.*)

1-6. Subject Matter and Problems.[1]—The subject matter of the book covers a wide range of questions in the fields of thermodynamics; chemistry, including combustion processes; mechanics of motion, including vibrations; strength of materials; and design of machine parts. All these questions are related to and often complicated by their application to internal-combustion engines.

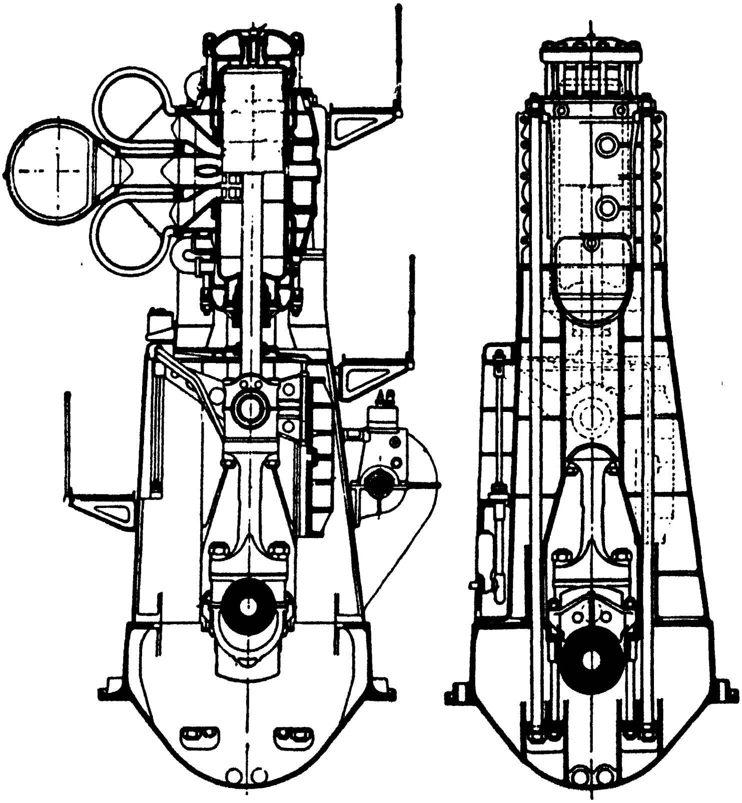

FIG. 1-20.—Double-acting mechanical-injection two-stroke oil engine.

Therefore, it is well to bear in mind that proper familiarity with the subject matter is not easily attained and certainly cannot be acquired through mere reading of the text without working out a certain minimum of typical and illustrative problems.

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[1] The following remarks belong rather to the preface to the book. However, the author's experience as a college instructor has taught him that many students do not read the preface; hence, he decided to put these remarks where they are more likely to be read and remembered.

As in practically all fields of engineering, the subject matter, while important in itself, is to a great extent only a background for the proper approach to various problems which may be encountered in practice.

Therefore, the working of the problems given in the book is very important. However, very little will be accomplished if a person glances through the problem, takes a formula which looks more or less suitable, picks out data to substitute in the formula, often not even paying attention to the units or dimensions which are required by the formula, runs a string of calculations on the slide rule, and gets some answer. The chances are that in about every other case the answer will not be right, and in most cases the person who went through such manipulations will have gained nothing.

A problem, in order to be of value, must be analyzed step by step and solved on its merits, not mechanically. The following procedure is recommended:

1. Read the problem attentively.
2. Write down the given data, with their dimensions, in a logical order.
3. List questions or results asked for.
4. Whenever practicable, draw a sketch to illustrate the problem.
5. Insert given data on the sketch.
6. Analyze the problem and note the kind of process or processes involved.
7. Note the principle which applies to each type of process.
8. Indicate the logical step-by-step solution, either by formulas given in the book or by setting up proper equations.
9. Calculate the results asked for.

If the analysis was correct and sufficient data are available, the solving of the problem should not present any difficulty.

If a problem is handled in the outlined manner, it will help the student to understand the underlying principles and the subject matter in general. A habit of working the problems in this manner will be a great asset for both classwork and future engineering practice at large.

CHAPTER 2

PRINCIPLES OF THERMODYNAMICS

2-1. Perfect Gases.—The application of the principles of thermodynamics to engine problems is considerably simplified if at first the working substances are considered to be perfect gases.

The three main variables of a gas are absolute pressure p, in pounds per square foot, volume V, in cubic feet, and temperature T, in degrees Fahrenheit absolute, more conveniently called Rankine degrees. If t is the temperature in degrees Fahrenheit,

$$T = t + 460 \tag{2-1}$$

Characteristic Equation.—For a perfect gas, which obeys the two fundamental laws of Boyle and Charles, the relation of the three variables can be given in the form of the characteristic equation

$$pV = WRT \tag{2-2}$$

where W is the weight in pounds and R is the gas constant in foot-pounds per pound per degree Rankine.

For 1 lb of gas the characteristic equation becomes

$$pv = RT \tag{2-3}$$

Mol System.—Computations where both weights and volumes of different substances are involved are considerably simplified by introducing the term *mol.* A mol means M lb of a substance where M is its molecular weight. For instance, 1 mol O_2 means 32 lb oxygen, 1 mol C_6H_6 (benzene) means 78 lb of this liquid. The specific volumes of solids and liquids are so small compared with the specific volumes of gases that in most technical computations they can be neglected. Avogadro's law states that equal volumes of any gases having the same pressure and temperature contain equal numbers of molecules. Therefore the ratio of weights of equal volumes of any two gases at the same pressure and temperature is equal to the ratio of their molecular weights.

Consequently, at a given pressure and temperature the volumes of 1 mol of all gases are the same. In technical measurements of gases a pressure of 14.7 psi and a temperature of 60 F as adopted by the SAE are used as *standard conditions.* For these standard conditions the volume of 1 mol of any gas is 379 cu ft and the specific volume v, or the volume of 1 lb, is

$$v = 379/M \tag{2-4}$$

This value is obtained by using the corresponding values for p and T and for the gas constant the expression

$$R = 1545/M \tag{2-5}$$

$1545(= MR)$ is called the *universal gas constant.* Thus the characteristic equation for n mols becomes

$$pV = 1545nT \tag{2-6}$$

Specific weight is the weight of 1 unit of volume. For gases and vapors the specific weight is expressed in pounds per cubic foot. Specific weight w is the reciprocal of the specific volume v and in accordance with equation (2-4) is proportional to the molecular weight M. Thus for any two gases the following relations exist:

$$w_1/w_2 = v_2/v_1 = M_1/M_2 \tag{2-7}$$

Specific Heat.—The heat required to raise the temperature of 1 pound of a substance 1 degree under certain external conditions is called the *thermal capacity* of the substance. The average thermal capacity per degree Fahrenheit of 1 lb water between 32 and 212 F is called 1 British thermal unit (Btu), and the thermal capacity expressed in Btu per pound per degree Fahrenheit is called *specific heat.*

The two very important conditions at which temperature changes of gases occur are *constant pressure* and *constant volume,* and the respective specific heats are designated c_p and c_v. The ratio of these specific heats is designated

$$k = c_p/c_v \tag{2-8}$$

and their difference is

$$c_p - c_v = AR \tag{2-9}$$

where $A = 1/778$ is the heat equivalent of work.

In computations where only comparative figures are sought, or at low temperatures, specific heats may be assumed to be constant. When greater accuracy is required or in dealing with higher temperatures, the influence of temperature must be taken into consideration.

For 1 mol the specific heats, also called *molar heat capacities*, are

$$\gamma_p = Mc_p \qquad \text{and} \qquad \gamma_v = Mc_v \tag{2-10}$$

Table 2-1 gives equations for finding instantaneous molar specific heats γ_p for various gases at constant pressure. The molar heat at constant volume is, substituting in equation (2-9) expression (2-5) for R and using the designations (2-10),

$$\gamma_v = \gamma_p - 1.986 \tag{2-11}$$

This means that the ratio of specific heats k also changes with temperature. The specific heats per pound of gas are found by dividing γ_p and γ_v by the molecular weight of the gas.

Gas Mixtures.—If several gases or vapors are contained in a closed space, then the pressures which the constituents would have, if each occupied the total space alone, are called *partial pressures.* If the partial pressures of the constituent gases are $p_1, p_2, p_3 \ldots$ and the total pressure of the mixture is p, then, according to Dalton's law,

$$p = p_1 + p_2 + p_3 \cdots \tag{2-12}$$

This law holds true if the gases or vapors in a mixture have no chemical affinity.

If the parts of the constituents by volume are $r_{v1}, r_{v2}, r_{v3} \ldots$ and the parts by weight are $r_{w1}, r_{w2}, r_{w3} \ldots$, where

$$r_{v1} + r_{v2} + r_{v3} \cdots = 1; \qquad r_{w1} + r_{w2} + r_{w3} \cdots = 1$$

then the partial pressures can be expressed as

$$p_1 = pr_{v1}; \qquad p_2 = pr_{v2}; \qquad p_3 = pr_{v3} \tag{2-13}$$

On the other hand, if the specific volume of the mixture is v and the temperature is T, then, from equation (2-2),

$$p_1 = r_{w1}R_1T/v; \qquad p_2 = r_{w2}R_2T/v; \qquad p_3 = r_{w3}R_3T/v \tag{2-14}$$

The sum of equations (2-14) gives in connection with equation (2-12)

$$p = \Sigma(r_wR)T/v \tag{2-15}$$

Comparing expression (2-15) with the characteristic equation for 1 lb of mixture,

$$pv = R_m T$$

gives for the gas constant of the mixture

$$R_m = \Sigma(r_w R) \qquad (2\text{-}16)$$

Similarly the specific heats of a mixture may be expressed as

$$c_{pm} = \Sigma(r_w c_p) \qquad (2\text{-}17)$$
$$c_{vm} = \Sigma(r_w c_v) \qquad (2\text{-}18)$$

Gas analysis is usually given in volumetric percentages or fractions. The change to fractions by weight, or vice versa, can be made using the evident relationships

$$r_{wi} = r_{vi} M_i / \Sigma(r_v M) \qquad (2\text{-}19)$$

and

$$r_{vi} = (r_{wi}/M_i)/\Sigma(r_w/M) \qquad (2\text{-}20)$$

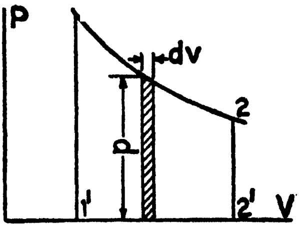

Fig. 2-1.—Pressure-volume diagram.

2-2. Forms of Energy.—Of the many forms in which energy can appear, only those encountered in internal-combustion engines will be mentioned.

1. *Mechanical Work.*—When a gas expands, it is capable of doing work equal to the product of pressure and change of volume

$$dW_k = p\,dV \qquad (2\text{-}21)$$

If a gas expands from condition 1 to condition 2, Fig. 2-1, the work W_k is represented by the area 1-2-2′-1′-1, and

$$W_k = \int_1^2 p\,dV \qquad (2\text{-}22)$$

If dV is positive, the gas is expanding and work is positive or is done by the gas; if dV is negative, the gas is being compressed and the negative work is done on the gas by an outside source.

Mechanical-work values are *absolute* values of energy which is actually transferred from a working fluid to an engine, or vice versa.

2. *Flow Work.*—Energy entering or leaving a process owing to the flow of the working fluid to or from the process is called

flow work. Numerically it is equivalent to the work required of a piston mechanism in moving the fluid past a given section in a pipe. This work is equal to pV, the product of the pressure and of the volume of the fluid at the given section. Flow work is not energy in the fluid but energy transferred into and out of the process by the fluid. Flow work represents an *absolute* value of energy.

Example 2-1.—Determine the flow work per mol of exhaust gases flowing through an exhaust pipe; the back pressure is 1.5 psi and the gas temperature is 1300 F.

The flow work is pV, which is equal to the left side of equation (2-6); therefore

$$pV = 1545(1300 + 460) = 2{,}720{,}000 \text{ ft-lb/mol}$$

In heat units

$$ApV = 2{,}720{,}000/778 = 3495 \text{ Btu/mol}$$

3. *Heat Q* is energy transferred to or from a substance owing to temperature difference. Heat transfer may be evaluated from the specific heat c and the temperature change

$$Q = \int_{T_1}^{T_2} Wc\,dT \tag{2-23}$$

where, for the most common changes, at constant volume or at constant pressure, c is a function of T as given in Table 2-1.

Heat represents an *absolute* value of energy.

If the specific heat is assumed constant, integrating equation (2-23) gives

$$Q = Wc(T_2 - T_1) = Wc(t_2 - t_1) \tag{2-24}$$

4. *Kinetic energy* K_i is energy of a substance due to its motion and is expressed as

$$K_i = Wv^2/2g \tag{2-25}$$

where $W/g = m$ is mass, lb-sec^2/ft,
v is velocity, fps.

Kinetic energy is a *relative* value, usually referred to the velocity of the earth surface as representing zero velocity.

Example 2-2.—Determine the kinetic energy per mol of exhaust gases in an exhaust line. The gas velocity is 600 fps; the apparent molecular weight of the gases is 28.7.

$$K_i = 28.7 \times 600^2/2 \times 32.2 = 160{,}500 \text{ ft-lb/mol or } 206 \text{ Btu/mol}$$

TABLE 2-1.—SPECIFIC HEAT EQUATIONS

Gas or vapor	Symbol	Molecular weight	Gas const. R	Molecular heat capacity γ_p at constant pressure Btu/lb-mol-°R	Limit of validity °R
Hydrogen	H_2	2.016	767.0	$5.76 + 5.76\times10^{-4}T + 20T^{-0.5}$	4000[a]
Hydroxide	OH	17.01	90.9	$7.401 - 0.7306\times10^{-3}T + 0.386\times10^{-6}T^2$	1620[b]
over 1620 R	......	....	....	$10.151 - 8217T^{-1} + 5.67\times10^6T^{-2}$	5940[b]
Steam	H_2O	18.02	85.8	$19.86 - 597T^{-0.5} + 7500T^{-1}$	5400[a]
Carbon monoxide	CO	28.0	55.2	$9.46 - 3.29\times10^3T^{-1} + 1.07\times10^6T^{-2}$	9000[b]
Nitrogen	N_2	28.02	55.2	$9.47 - 3.47\times10^3T^{-1} + 1.16\times10^6T^{-2}$	5000[a]
Nitric oxide	NO	30.01	51.5	$6.550 + 0.889\times10^{-3}T$	1620[b]
over 1620 R	......	....	....	$9.406 - 2.293\times10^3T^{-1}$	5940[b]
Oxygen	O_2	32.0	48.3	$11.515 - 172T^{-0.5} + 1530T^{-1}$	5000[a]
Carbon dioxide	CO_2	44.0	35.1	$16.2 - 6.53\times10^3T^{-1} + 1.41\times10^6T^{-2}$	6300[a]
Methane	CH_4	16.03	96.4	$4.22 + 8.211\times10^{-3}T$	1800[b]
over 1800 R	......	....	....	$27.0 - 14{,}400T^{-1}$	5940[b]
Acetylene	C_2H_2	26.02	59.4	$7.51 + 0.0062T$	1100[c]
Ethylene	C_2H_4	28.03	55.1	$4.23 + 0.0118T$	1100[c]
Ethane	C_2H_6	30.05	51.5	$4.01 + 0.0164T$	1100[c]
Propylene	C_3H_6	42.05	36.7	$4.09 + 0.0207T$	1100[c]
Propane	C_3H_8	44.06	35.1	$4.09 + 0.0240T$	1100[c]
Butane	C_4H_{10}	58.08	26.6	$4.60 + 0.0314T$	1100[c]
Pentane	C_5H_{12}	72.09	21.4	$5.11 + 0.0386T$	1100[c]
Benzene	C_6H_6	78.05	19.8	$6.16 + 0.0292T$	1100[c]
Hexane	C_6H_{14}	86.11	17.85	$5.62 + 0.0457T$	1100[c]
Heptane	C_7H_{16}	100.12	15.43	$6.13 + 0.0529T$	1100[c]
Octane	C_8H_{18}	114.14	13.55	$14.40 + 0.0533T$	1440[b]
Decane	$C_{10}H_{22}$	142.17	10.87	$7.66 + 0.0746T$	1100[c]
Dodecane	$C_{12}H_{26}$	170.2	9.08	$8.68 + 0.0889T$	1100[c]
Hexadecadiene	$C_{16}H_{30}$	222.23	6.95	$10.72 + 0.2337T$	1100[c]

[a] SWEIGERT, R. L., and BEARDSLEY, M. W., Empirical Specific Heat Equations Based upon Spectroscopic Data, *Georgia State Eng. Exp. Sta. Bull.* 2, 1938.

[b] SCHWARZ, C., Die spezifische Wärmen der Gase als Hilfswerte zur Berechnung der Gleichgewichten, *Arch. Eisenhüttenw.*, vol. 9, p. 389, 1936.

[c] EDMISTER, W. C., Thermodynamic Properties of Hydrocarbons, *J. Ind. Eng. Chem.*. vol. 30, p. 354, March, 1938.

TABLE 2-2.—INTERNAL ENERGY OF COMBUSTION GASES
(Btu per lb-mol above 520 R)*

Temp., °R	O_2	N_2	Air	H_2O	CO_2	H_2	CO	OH	NO	Mona-tomic O,H,A	C_8H_{18}	$C_{12}H_{26}$	$ApV = ART$	Temp., °R
520	0	0	0	0	0	0	0	0	0	0	0	0	1,033	520
560	200	196	196	244	280	193	196	202	207	119	1,648	5,297	1,112	560
600	402	395	395	490	570	390	396	407	418	238	3,381	10,967	1,192	600
700	920	896	897	1,110	1,320	887	896	914	943	537	8,088	26,770	1,390	700
800	1,449	1,399	1,403	1,734	2,120	1,386	1,402	1,422	1,468	834	13,328	44,900	1,589	800
900	1,989	1,905	1,915	2,366	2,965	1,886	1,913	1,930	1,994	1,132	19,099	65,410	1,787	900
1000	2,539	2,416	2,431	3,009	3,852	2,387	2,430	2,439	2,522	1,430	25,403	88,210	1,986	1000
1100	3,101	2,934	2,957	3,666	4,778	2,889	2,954	2,948	3,053	1,728	32,241	113,500	2,185	1100
1200	3,675	3,461	3,492	4,339	5,736	3,393	3,485	3,458	3,596	2,026	39,612	140,800	2,383	1200
1300	4,262	3,996	4,036	5,030	6,721	3,899	4,026	3,968	4,154	2,324	47,516	170,600	2,582	1300
1400	4,861	4,539	4,587	5,740	7,731	4,406	4,580	4,480	4,730	2,622	55,953	202,700	2,780	1400
1500	5,472	5,091	5,149	6,488	8,764	4,916	5,145	4,996	5,321	2,920	64,923	....	2,979	1500
1600	6,092	5,652	5,720	7,212	9,819	5,429	5,720	5,516	5,922	3,218	74,425	....	3,178	1600
1700	6,718	6,224	6,301	7,970	10,896	5,945	6,305	6,041	6,532	3,516	84,461	....	3,376	1700
1800	7,349	6,805	6,889	8,741	11,993	6,464	6,899	6,571	7,150	3,813	95,030	....	3,575	1800
1900	7,985	7,393	7,485	9,526	13,105	6,988	7,501	7,106	7,730	4,111	106,132	....	3,773	1900
2000	8,629	7,989	8,067	10,327	14,230	7,517	8,109	7,646	8,399	4,409	117,767	....	3,972	2000
2100	9,279	8,592	8,698	11,146	15,368	8,053	8,722	8,191	9,029	4,707	....	....	4,171	2100
2200	9,934	9,203	9,314	11,983	16,518	8,597	9,339	8,741	9,663	5,005	....	....	4,369	2200
2300	10,592	9,817	9,934	12,835	17,680	9,147	9,961	9,299	10,302	5,303	....	....	4,568	2300
2400	11,252	10,435	10,558	13,700	18,852	9,703	10,588	9,867	10,946	5,601	....	....	4,766	2400
2500	11,916	11,056	11,185	14,578	20,033	10,263	11,220	10,445	11,595	5,899	....	....	4,965	2500
2600	12,584	11,682	11,817	15,469	21,222	10,827	11,857	11,039	12,249	6,197	....	....	5,164	2600
2700	13,257	12,313	12,453	16,372	22,419	11,396	12,499	11,628	12,906	6,495	....	....	5,362	2700
2800	13,937	12,949	13,095	17,288	23,624	11,970	13,144	12,223	13,566	6,792	....	....	5,561	2800
2900	14,622	13,590	13,742	18,217	24,836	12,549	13,792	12,823	14,228	7,090	....	....	5,759	2900

3000	15,309	14,236	14,394	19,160	26,055	13,133	14,443	13,428	14,892	7,388	....	....	5,958	3000
3100	16,001	14,888	15,051	20,177	27,281	13,723	15,097	14,037	15,559	7,686	....	....	6,157	3100
3200	16,693	15,543	15,710	21,086	28,513	14,319	15,754	14,649	16,229	7,984	....	....	6,355	3200
3300	17,386	16,199	16,369	22,066	29,750	14,921	16,414	15,264	16,902	8,282	....	....	6,554	3300
3400	18,080	16,855	17,030	23,057	30,991	15,529	17,078	15,882	17,577	8,580	....	....	6,752	3400
3500	18,776	17,512	17,692	24,057	32,237	16,143	17,744	16,504	18,254	8,878	....	....	6,951	3500
3600	19,475	18,171	18,356	25,067	33,487	16,762	18,412	17,131	18,931	9,176	....	....	7,150	3600
3700	20,179	18,833	19,022	26,085	34,741	17,385	19,082	17,762	19,610	9,474	....	....	7,348	3700
3800	20,887	19,496	19,691	27,110	35,998	18,011	19,755	18,397	20,291	9,771	....	....	7,547	3800
3900	21,598	20,162	20,363	28,141	37,258	18,641	20,430	19,036	20,973	10,069	....	....	7,745	3900
4000	22,314	20,830	21,037	29,178	38,522	19,274	21,107	19,678	21,657	10,367	....	....	7,944	4000
4100	23,034	21,500	21,714	30,221	39,791	19,911	21,784	20,324	22,343	10,665	....	....	8,143	4100
4200	23,757	22,172	22,393	31,270	41,064	20,552	22,462	20,975	23,030	10,963	....	....	8,341	4200
4300	24,482	22,845	23,073	32,326	42,341	21,197	23,140	21,631	23,719	11,261	....	....	8,540	4300
4400	25,209	23,519	23,755	33,389	43,622	21,845	23,819	22,292	24,409	11,558	....	....	8,738	4400
4500	25,938	24,194	24,437	34,459	44,906	22,497	24,499	22,957	25,100	11,856	....	....	8,937	4500
4600	26,668	24,869	25,120	35,535	46,193	23,154	25,179	23,626	25,792	12,154	....	....	9,136	4600
4700	27,401	25,546	25,805	36,616	47,483	23,816	25,860	24,297	26,485	12,452	....	....	9,334	4700
4800	28,136	26,224	26,491	37,701	48,775	24,480	26,542	24,969	27,179	12,750	....	....	9,533	4800
4900	28,874	26,905	27,180	38,791	50,069	25,148	27,226	25,643	27,874	13,048	....	....	9,731	4900
5000	29,616	27,589	27,872	39,885	51,365	25,819	27,912	26,319	28,570	13,346	....	....	9,930	5000
5100	30,361	28,275	28,566	40,983	52,663	26,492	28,600	26,997	29,267	13,644	....	....	10,129	5100
5200	31,106	28,961	29,262	42,084	53,963	27,166	29,289	27,677	29,966	13,942	....	....	10,327	5200
5300	31,857	29,648	29,958	43,187	55,265	27,842	29,980	28,360	30,666	14,240	....	....	10,526	5300
5400	32,607	30,337	30,655	44,293	56,569	28,519	30,674	29,045	31,367	14,537	....	....	10,724	5400
5500	33,386	31,026	31,353	45,402	57,875	29,298	31,369	29,731	32,069	14,835	....	....	10,923	5500

---

* Based on data of R. L. Hershey, J. E. Eberhardt, and H. C. Hottel, Thermodynamic Properties of the Working Fluid in Internal-combustion Engines, *SAE Journal*, vol. 39, No. 4 p. 423, October, 1936; L. C. Lichty, "Internal-combustion Engines," 5th ed., pp. 582–583, McGraw-Hill Book Company, Inc.. New York, 1936.

which is only about 0.1 per cent of the energy liberated during combustion of 1 mol of fuel.

5. *Internal Energy.*—The energy of gas due to its molecular motion is called *internal energy* U. For a perfect gas, U depends only upon its temperature. The infinitesimal change of internal energy per mol of gas is

$$dU = \gamma_v \, dT$$

or

$$dU = Mc_v \, dT \tag{2-26}$$

Usually only the change of internal energy during a process is of interest.

Integrating expression (2-26) from T_1 to T_2 gives this change as

$$U_2 - U_1 = M \int_{T_1}^{T_2} c_v \, dT \tag{2-27}$$

The zero of internal energy is taken arbitrarily at a certain temperature. For sake of convenience, values of U, computed from the standard temperature of 60 F or 520 R by equation (2-27) and using data from Table 2-1 for the molar specific heats, are given in Table 2-2.

Internal energy thus computed represents a *relative* value.

Example 2-3.—Determine the value of the internal energy of 1 mol of methane, CH_4, at a temperature of 77 F.

In equation (2-27),

$$T_1 = 520 \text{ R} \qquad \text{and} \qquad T_2 = 460 + 77 = 537 \text{ R}$$

From Table 2-1, for CH_4, with equation (2-11)

$$\gamma_v = 4.22 + 0.008211T - 1.986 = 2.234 + 0.008211T$$

by equation (2-27)

$$\begin{aligned} U_{CH_4} &= \int_{520}^{537} (2.234 + 0.008211T) \, dT \\ &= 2.234(537 - 520) + 0.5 \times 0.008211(537^2 - 520^2) = 112 \text{ Btu/mol} \end{aligned}$$

6. ***Enthalpy.***—An infinitesimal quantity of heat added to 1 mol of a gas at constant pressure can be presented as

$$dH = Mc_p \, dT \tag{2-28}$$

Expression (2-28) integrated from T_1 to T_2 gives

$$H_2 - H_1 = M \int_{T_1}^{T_2} c_p \, dT \tag{2-29}$$

The change of enthalpy H represents the amount of heat which must be added to raise the temperature of the gas from T_1 to T_2 and also to do the work of expansion, because the volume of a gas at constant pressure increases with the increase of temperature.

Another definition states that enthalpy is the sum of the internal energy and of the term pV, which represents flow work if the substance is flowing, expressed in Btu,

$$H = U + ApV \tag{2-30}$$

where with equations (2-6), (2-9), and (2-11) for 1 mol, $n = 1$,

$$H = U + 1.986T \tag{2-31}$$

Since U has a relative value, enthalpy also represents a *relative* value.

7. *Chemical energy C* is energy stored in the electron and the bonding arrangement of the atoms of a molecule. Changes in this arrangement due to combination with other atoms or dissociation result in the liberation or absorption of energy. Chemical energy is computed from the heats of reaction of combustion and from values of internal energy, or enthalpy, of the substance before and after the reaction. Therefore chemical energy represents a *relative* value.

Difference of Energy Terms.—Work and heat transfer depend upon the path of the process, and the plotting of the path is possible only when the working fluid is in a state of equilibrium throughout the process. Changes of other forms of energy, kinetic energy, internal energy, enthalpy, and chemical energy, can be computed from the knowledge of initial and final conditions.

Some of the energy forms, W_k, pV, and Q, are absolute in value, while others, K_i, U, H, and C, are relative and in some instances may be based upon different reference planes. These differences must be taken into account. For processes in which the amount of working substance does not change, the differences of values of energy at the beginning and at the end of the process, such as given by equation (2-27) or (2-29), eliminate the reference-plane values and give absolute energy changes.

2-3. Availability of Energy. *Fundamental Equation.*—According to the first law of thermodynamics, in a nonflow process heat added or rejected is equal to the sum of the change of internal energy and of the external work done,

$$Q = (U_2 - U_1) + AW_k \tag{2-32}$$

In differential form, equation (2-27) in connection with expression (2-21) can be written as

$$dQ = dU + Ap\,dV \tag{2-33}$$

Entropy is a convenient mathematical concept. Sometimes entropy is defined as that function of the working substance which remains constant during an adiabatic process. The mathematical expression for entropy is

$$dS = dQ/T \tag{2-34}$$

or

$$S = \int dQ/T \tag{2-35}$$

To become familiar with entropy one may consider it as one of the variables which, with absolute temperature taken as the second variable, form a special system of coordinates. In this system an area enclosed by lines representing the change of state of the working substance represents heat in the same manner as an area enclosed by lines representing the change of pressure and volume represents work in a pressure-volume system of coordinations.

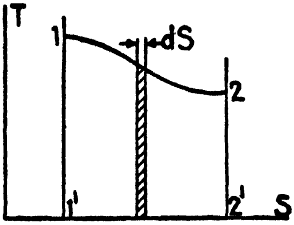

FIG. 2-2.—Temperature-entropy diagram.

Entropy is designated by S when referred to W lb of a substance and s when referred to 1 lb. Thus dS, Fig. 2-2, represents a change of entropy corresponding to a change of volume dV, Fig. 2-1, and its product by T corresponding to pressure P represents a change of heat $dQ = T\,dS$.

For 1 lb, in connection with equation (2-33),

$$T\,ds = du + Ap\,dv \tag{2-36}$$

In the same manner, area 1-2-2′-1′-1, Fig. 2-2, gives the heat q added or rejected during a change of state from 1 to 2 corresponding to the work done from 1 to 2, Fig. 2-1,

$$\int_1^2 T\,ds = \int_1^2 c_v\,dT + A\int_1^2 p\,dv \tag{2-37}$$

In a cycle the change of state becomes a closed curve in both the p-V and T-s diagrams, and the change of internal energy becomes zero; therefore areas enclosed by the curves become equivalent or equal if the scales are so selected as to have 1 Btu = 778 ft-lb.

Figure 2-3 shows a cycle in T-S coordinates. Points 1 and 2 are the points with the lowest and the highest entropies, respectively. On curve 1-*a*-2 the working substance receives Q_1 Btu of heat, represented by the area 1-*a*-2-2′-1′-1; on curve 2-*b*-1 the working substance loses Q_2 Btu of heat—area 2-*b*-1-1′-2′-2. The difference of heats $Q_1 - Q_2$ represented by the area 1-*a*-2-*b*-1 is transformed into work W_k of the cycle, and the ratio is the efficiency of the cycle

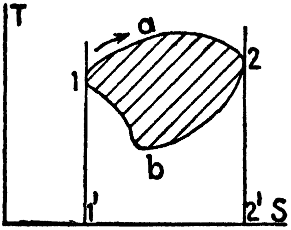

FIG. 2-3.—Entropy diagram of a cycle.

$$\eta = (Q_1 - Q_2)/Q_1 = AW_k/Q_1 \tag{2-38}$$

Determining ds from expression (2-36), eliminating u and p by means of equations (2-26) and (2-3), gives, after taking the integral, for lines of v = const.,

$$s = \int_1^2 c_v \frac{dT}{T} + AR \log_e v + C_1 \tag{2-39}$$

Eliminating v with the aid of equation (2-3) gives for p = const.

$$s = \int_1^2 c_p \frac{dT}{T} - AR \log_e p + C_2 \tag{2-40}$$

Equations (2-39) and (2-40) show that entropy is of the order of specific heats and is expressed in Btu per pound per degree Rankine.

Available Energy.—That part of heat added to a system which can be transformed into work is called *available energy*. A system performing the Carnot or the ideal cycle transforms into work the largest amount of heat, and this amount of heat is the available energy under the given conditions. The ratio of available energy to the total heat added is called the *availability of energy*, and, in the case of a continuous performance as in a cycle, this ratio is called *efficiency*.

The efficiency of a Carnot cycle, whose source of heat has a temperature of T_1 and whose refrigerator has a temperature of T_0, is

$$\eta_c = (T_1 - T_0)/T_1 \tag{2-41}$$

and is the highest which can be attained between these temperatures. If a system obtains Q Btu of heat at a temperature T_1 and the lowest attainable temperature for a refrigerator is T_0, then the available heat energy Q_a is

$$Q_a = \eta_c Q = Q(T_1 - T_0)/T_1 \tag{2-42}$$

The unavailable part Q_u is

$$Q_u = Q - Q_a = QT_0/T_1 \tag{2-43}$$

If heat is added not isothermically and the temperature of a body is raised from T_1 to T_2, the amount of unavailable heat may be determined from the general equation

$$Q_u = \int dQT_0/T \tag{2-44}$$

Substituting for heat added the general expression

$$dQ = Wc\,dT \tag{2-45}$$

and taking the definite integral gives

$$Q_u = \int_{T_1}^{T_2} T_0 Wc\,dT/T = Wc\,T_0 \log_e (T_2/T_1) \tag{2-46}$$

The expression (2-46) is general and can be used for any change of state, provided that c is constant. For heat added at constant volume, $c = c_v$; for constant pressure $c = c_p$; for any polytropic change $c = c_n$.

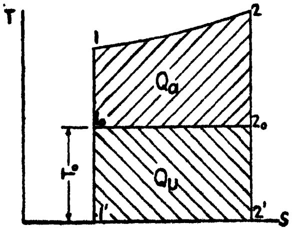

FIG. 2-4.—Available and unavailable energies.

The relation between the total heat change Q, the available energy Q_a, and unavailable energy Q_u can be shown graphically, Fig. 2-4: during a change of state, represented by the curve 1-2, Q is equal to the area 1-2-2′-1′-1; Q_u is equal to the area cut off by the refrigerator temperature T_0, area 1_0-2_0-2′-1′-1_0; and the difference, area 1-2-2_0-1_0-1, is the available energy Q_a.

Figure 2-4 shows why it is much easier to compute Q_u than Q_a and that $\log_e (T_2/T_1) = S_2 - S_1$, expression (2-46). That part of the heat which can be transformed into work under actual conditions is always smaller than the available energy because of different losses. One of the main problems of the theory of internal-combustion engines is to determine how to increase the availability of heat and through it the efficiency of the engine cycle.

2-4. Polytropic changes are presented by the equation

$$pV^n = C \tag{2-47}$$

where, depending upon the actual value of the exponent n, the equation may cover an infinite number of changes. Any change of state, such as shown on an indicator card, may be expressed by this equation (2-47).

General Equations.—Designating all quantities at the beginning of the polytropic change by subscripts 1 and at the end by subscripts 2, the relation between initial and final pressures and volumes can be written directly from equation (2-47)

$$p_2/p_1 = (V_1/V_2)^n \tag{2-48}$$

Equations (2-47) and (2-2) combined to eliminate p give another form of polytropic equation

$$TV^{n-1} = C \tag{2-49}$$

which gives the relation between temperatures and volumes

$$T_2/T_1 = (V_1/V_2)^{n-1} \tag{2-50}$$

Equations (2-47) and (2-2) combined to eliminate V give the third form of polytropic equation

$$p^{\frac{n-1}{n}}/T = C \tag{2-51}$$

which gives the relation between temperatures and pressures

$$T_2/T_1 = (p_2/p_1)^{(n-1)/n} \tag{2-52}$$

Work W_k done during polytropic expansion or absorbed during polytropic compression can be found from equations (2-22) and (2-47)

$$W_k = (p_1V_1 - p_2V_2)/(n - 1) \tag{2-53}$$

Using equation (2-2), this expression may be presented also as

$$W_k = WR(T_1 - T_2)/(n - 1) \tag{2-54}$$

If W and R are not constant but change, as during combustion in an engine with a gradual injection and combustion of the fuel, equation (2-54) may be presented as

$$W_k = (W_1R_1T_1 - W_2R_2T_2)/(n - 1) \tag{2-54a}$$

However, in this case equation (2-53) is more convenient.

The amount of heat added or rejected during a polytropic change can be computed using expressions (2-27) and (2-54) in equation (2-32), which gives

$$Q = Wc_v \frac{n - k}{n - 1} (T_2 - T_1) \tag{2-55}$$

Expression $c_v(n - k)/(n - 1)$ is designated as c_n and is called the *specific heat of a polytropic change,*

$$c_n = c_v(n - k)/(n - 1) \tag{2-56}$$

This gives for heat added or rejected

$$Q = Wc_n(T_2 - T_1) \tag{2-57}$$

c_n can be either positive or negative, depending upon the numerical value of the exponent n. However, even if c_n is negative, Q may be positive, meaning that heat is added to the system, namely, if simultaneously T_2 is smaller than T_1.

The amount of unavailable energy can be computed from expression (2-46) using c_n for c.

In order to show how curves and lines found in indicator cards can be considered special cases of polytropic curves, the following cases will be analyzed:

Isochore: V = const. This is a polytropic line with $n = \infty$, which is obvious if equation (2-47) is presented as $p^{1/n}V = C$.

Isobar: p = const. This is obtained directly when $n = 0$.

Isothermal for a gas is a curve

$$pV = C \tag{2-58}$$

meaning that $n = 1$.

Adiabatic is obviously obtained with $n = k$,

$$pV^k = C \tag{2-59}$$

Graphic Presentation.—Curves with exponents n having values from 0 to ∞ have the same nature as an isothermic or adiabatic curve—they are convex to the lines of coordinates, Fig. 2-5.

This investigation continued to negative values of n shows that $n = -1$ represents inclined straight lines, the angle of slope being determined by the value of the constant C, equation (2-47). Lines with n between 0 and -1 are curves concave to the axis of abscissas, Fig. 2-5; lines with n between -1 and $-\infty$ are curves concave to the axis of ordinates. Thus, by proper selection of n and C, any regular curve can be expressed by one or several equations (2-47).

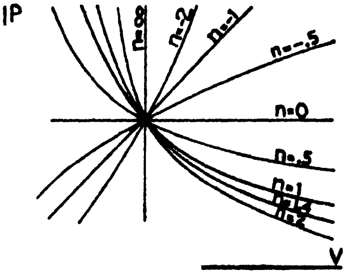

FIG. 2-5.—Polytropic curves.

Special Cases.—Isochore: $V = C$. With $n = \infty$ the relation between T and p, equation (2-52), becomes

$$T_2/T_1 = p_2/p_1 \tag{2-60}$$

Equation (2-53) gives $W_k = 0$, and equation (2-57) becomes, with $c_n = c_v$,

$$Q = Wc_v(T_2 - T_1) \tag{2-61}$$

Isobar: $p = C$. With $n = 0$ the relation between T and V, equation (2-50), becomes

$$T_2/T_1 = V_2/V_1 \tag{2-62}$$

Expression for work W_k, equation (2-53), becomes

$$W_k = p(V_2 - V_1) \tag{2-63}$$

and heat added per pound, equation (2-57) becomes, with $c_n = c_p$,

$$Q = Wc_p(T_2 - T_1) \tag{2-64}$$

Isothermal Process: $T = C$. With $n = 1$ the work W_k can be determined from the general equation (2-22), substituting p from equation (2-58),

$$W_k = pV \log_e (V_2/V_1) \tag{2-65}$$

Heat added can be calculated from the general equation (2-32), which, in connection with equation (2-27), gives

$$Q = WApV \log_e (V_2/V_1) = WART \log_e (V_2/V_1) \tag{2-66}$$

Adiabatic: $Q = 0$; $n = k$. The relation between p and V is given by equation (2-59). All equations from (2-48) to (2-54) expressing the relations between p, V, T, and W_k may be used simply substituting k for n.

2-5. Graphic Methods.—A polytropic process expressed by the equation (2-47) can be drawn as a curve by computing a number of pairs of p and V, plotting these points, and connecting them by a smooth curve.

If the value of n does not differ much from k, as in the case of compression and expansion lines in internal-combustion-engine indicator diagrams, the curve can be constructed graphically if one point, one pair of p and V, and the value of n are known.

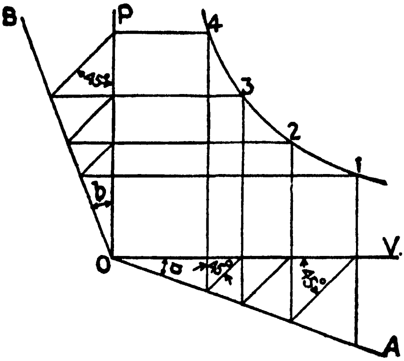

FIG. 2-6.—Construction of a polytropic curve.

Figure 2-6 shows the procedure due to Professor Brauer: A line O-A is drawn at an arbitrary angle a with the axis of abscissas; a second line O-B is drawn at an angle b with the axis of ordinates. The angle b is determined from the equation

$$\tan b = (1 + \tan a)^n - 1 \qquad (2\text{-}67)$$

Starting from the given point 1, zigzag lines are drawn as shown on Fig. 2-6 and the intersections 2, 3, 4, . . . are connected by a smooth curve.

The smaller the angle a, the more numerous the intermediate points. However, the whole construction must be made very carefully, for any inaccuracies accumulate in the next points. A convenient value for $\tan a$ is 0.25.

Determination of Exponent.—The value of the exponent n of a polytropic curve can be found from expression (2-48) by means of a log-log slide rule.

With a graphic method, the axes of abscissas and of ordinates are drawn through the point at zero pressure and zero volume, Fig. 2-7. Area 1-2-3-4-1 represents work W_k of expansion from point 1 to 2 and, as found before,

$$W_k = (p_1V_1 - p_2V_2)/(n - 1) \qquad [2\text{-}53]$$

Area 1-2-5-6-1 represents work W_c performed between pressures p_1 and p_2 as in a cycle and can be expressed as

$$W_c = n(p_1V_1 - p_2V_2)/(n - 1) \qquad (2\text{-}68)$$

Dividing (2-68) by (2-53) gives

$$n = W_c/W_k \qquad (2\text{-}69)$$

These methods give the average value of n for the whole curve from 1 to 2. If n is not constant throughout the curve 1-2, the latter can be divided into two or more parts and the value of n found for these separate sections of the curve.

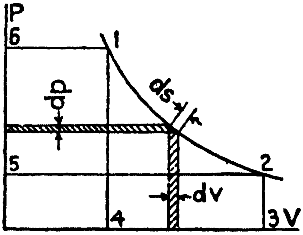

FIG. 2-7.—Determination of the exponent in polytropic curves.

A method of finding n for any single point of a curve may be explained as follows:[1] For an infinitesimal length ds of the polytropic curve, Fig. 2-7, expression (2-69) becomes

$$n = -V\,dp/p\,dV \qquad (2\text{-}70)$$

A tangent drawn to the curve through a point m, Fig. 2-8, until it intersects with the axis of abscissas at point b forms an angle a and

$$\tan a = -dp/dV$$

The subtangent

$$s = cb = \frac{cm}{\tan a} = -\frac{p}{dp/dV}$$

In connection with equation (2-70),

$$s = V/n \qquad (2\text{-}71)$$

A straight line e-e is drawn parallel to the line of abscissas at a distance selected as 1—one unit. Point b is projected on this line at point *d*; point *c* is the projection of point *m* on the line of abscissas. A straight line is then drawn through *d* and *c* until it intersects the line of ordinates at *f*, giving *o-f* = *n*.

---

[1] SCHWEITZER, P. H., *Penn. State Coll. Eng. Exp. Sta. Bull.* 35, 1926.

This can be proved from the similarity of triangles *c-b-d* and *c-o-f*.

Projecting the point *f* on the extension of line *c-m* and repeating this procedure for a number of points on the investigated polytropic curve gives a curve *n-n*, which shows the varying values of the exponent *n*.

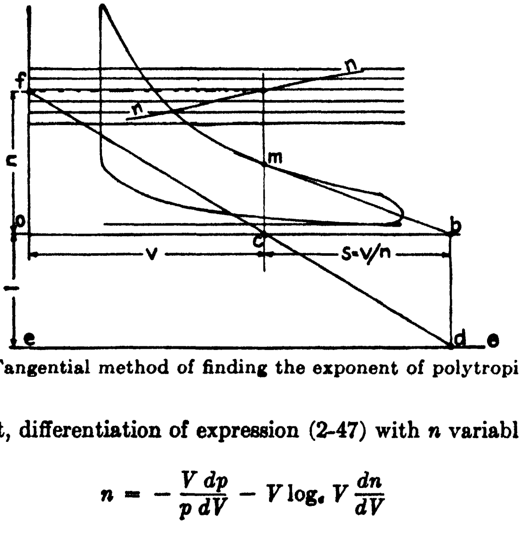

FIG. 2-8.—Tangential method of finding the exponent of polytropic curves.

To be exact, differentiation of expression (2-47) with *n* variable gives

$$n = -\frac{V\,dp}{p\,dV} - V \log_e V \frac{dn}{dV} \tag{2-72}$$

Expression (2-72) becomes identical with expression (2-70) if dn/dV is equal to zero, which is sufficiently close in regard to indicator diagrams taken from internal-combustion engines.

2-6. Problems.—1. Prove that the volume of 1 mol of any gas at standard conditions, $p = 14.7$ psia, $t = 60$ F, is 379 cu. ft.

2. Find the volume of 1 mol air at a pressure of 14.1 psia and a temperature of 42 F.

3. Find the gas constant and specific volume of methane, CH_4, at standard conditions.

4. Find the partial pressures of oxygen and nitrogen in atmospheric air, which contains by weight 23.1 per cent O_2, 76.9 per cent N_2.

5. Find the gas constant of air with the data given in problem 4.

6. Find the gas constant and specific heats c_p and c_v per pound at 100 F of a blast-furnace gas the volumetric analysis of which gave 3.0 per cent H_2, 27.1 per cent CO, 0.72 per cent CH_4, 0.38 per cent O_2, 9.9 per cent CO_2, and the balance N_2.

7. Find the gas constant and specific heats c_p and c_v per pound at 200 F of a natural gas which has the average composition, by volume, 84 per cent methane, 14 per cent ethane, and the balance nitrogen.

8. Find the temperature and pressure at the end of a polytropic compression of 2 cu ft air under standard conditions compressed to one-fifth of its original volume. The value of the exponent n is 1.32.

9. Find the work of compression and the heat rejected during the change of state given in problem 8.

10. One pound of air at a pressure of 14.0 psia and a temperature of 115 F is compressed until its pressure is doubled. Determine the final temperatures and volumes for three processes: (*a*) adiabatic, (*b*) isothermal, and (*c*) isochoric.

11. Fifty Btu of heat are added to a gas at a constant temperature of 600 F. How much of this heat is available with a refrigerator temperature of 68 F?

12. Twenty Btu are added to 0.4 lb water, whose initial temperature was 70 F. How much of this heat is available if the refrigerator temperature is 55 F?

13. A body H with a temperature of 1000 F, a mass of 5 lb, and a specific heat of 0.24 Btu/lb gives up heat to a cold body C having a temperature of 70 F, a mass of 4 lb, and a specific heat of 0.55. Determine Q_a and Q_u for each body and the net Q_a and Q_u for the isolated system containing bodies H and C if the refrigerator temperature is 52 F.

14. 1.5 cu ft of air is compressed to 0.3 cu ft; the initial pressure of 14.20 psia rises to 115 psi; the compression follows a polytropic curve; find its exponent.

15. Two pounds of air at 100 psia occupy 4 cu ft; the air is heated and expands at a constant pressure until its volume is increased 300 per cent; after that the air expands adiabatically to 50 psi, and finally its pressure is reduced isochorically to 15 psi. Determine the changes of available and unavailable energy and of the entropy between the starting and final conditions. Assume the refrigerator temperature to be 70 F, and use average specific heats for the temperature range involved.

16. Using the graphic method, draw an expansion curve with $n = 1.35$ for $V_1 = 150$ cu in., $p_1 = 100$ psig and $V_2 = 600$ cu in.; check by computing pressure for an intermediate volume $V = 400$ cu in.

17. With a radius of 5 in., draw an arc through points $x_1 = 1$ in., $y_1 = 1\frac{1}{2}$ in., $x_2 = 4$ in., and $y_2 = \frac{3}{8}$ in.; find by the graphic method of tangents the curve of instantaneous n for this arc.

CHAPTER 3

FUELS

3-1. Fuels Available.—Fuels used in internal-combustion engines come from all three groups—gaseous, liquid, and solid.

Gaseous fuels mostly used at present are natural gas, coke-oven gas, blast-furnace gas, and producer gas.

Liquid fuels are derived mostly from petroleum. The more important in this group are gasoline and fuel oil; next come gas oil and kerosene. A fuel not used much in the United States but having a potential future is alcohol, obtained either from grain or from wood.

All liquid fuels can be divided into two main groups: liquids that are vaporized and handled similarly to gas fuels—gasoline and alcohol are the main ones in this group, distillate and kerosene being used considerably less—and liquids that are injected into the combustion space—fuel oils of different characteristics, sometimes also kerosene or coal oil and gas oil.

Solid fuels are coal, chiefly anthracite and coke, up to lately used only indirectly, in gas producers. The problem of using coal directly by pulverizing it and burning in a compression-ignition engine gradually is being brought nearer to a practical solution by continuous research.[1] The main obstacle is the excessive wear of the cylinder liners and piston rings.[2]

Composition.—In all fuels the two basic combustible elements are carbon and hydrogen, encountered either separately or in combinations called *hydrocarbons.* At atmospheric temperature and pressure some of the hydrocarbons are gases, while some are liquids. The general chemical formula for all hydrocarbons is C_nH_m. The main types of hydrocarbons are paraffins, C_nH_{2n+2}, olefins, and naphthenes, C_nH_{2n}, diolefins, C_nH_{2n-2}, aromatics, C_nH_{2n-6}, and asphaltics, C_nH_{2n-4}. In different combinations of interest, as internal-combustion-engine fuel, *n* varies from 1 to about 26 and *m* from 2 to 54.

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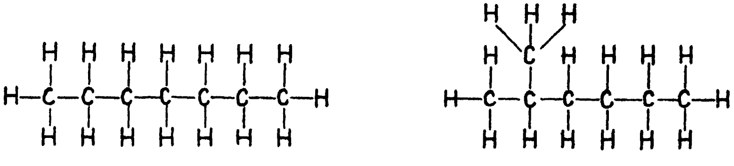
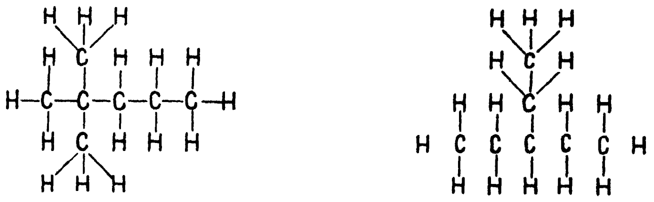
[1] *Power*, vol. 68, p. 136, July 24, 1928; p. 746, Nov. 6, 1928; *Z. Ver. deut. Ing.*, vol. 78, p. 1007, Aug. 25, 1934; Vol. 80, p. 269, Mar. 7, 1936; Spausta, Franz, "Treibstoffe für Verbrennungsmotoren," pp. 130–314, Julius Springer, Vienna, 1939.

[2] *Z. Ver. deut. Ing.*, vol. 80, p. 1099, Sept. 5, 1936; vol. 82, p. 1403, Dec. 3, 1938.



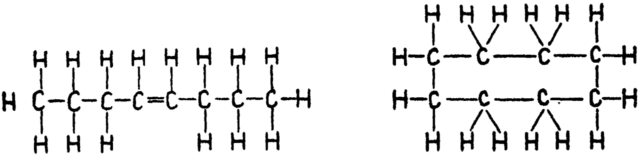
(a) Heptane, C_7H_{16} (b) Methyl Hexane (2)



(c) Dimethyl Pentane (2)(2) (d) Ethyl Pentane (3)

FIG. 3-1.—Isomers of a paraffin hydrocarbon (C_nH_{2n+2}).

The *paraffin* series in turn can be subdivided into *normal* paraffin hydrocarbons, which have a *straight* or *open-chain structure* with one bond between each atom, such as heptane, Fig. 3-1*a*, and *isomers*, which have the same number of C and H atoms

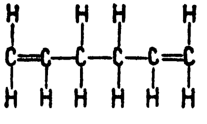


(a) Octene (4), C_8H_{16} (b) Cyclo-Octane, C_8H_{16}

FIG. 3-2.—Isomers of a naphthene hydrocarbon (C_nH_{2n}).

and the same molecular weight but a different structure, such as the three isomers of normal heptane, Fig. 3-1, *b*, *c*, and *d*. The numbers indicate the positions of the carbon atoms to which the methyl or ethyl groups are attached. More complicated paraffin hydrocarbons may have a greater number of isomers. The difference in the structure of the molecules results in different physical properties and reaction characteristics.

The *olefins*, C_nH_{2n}, are also *straight* or *open-chain* hydrocarbons but have two atoms of hydrogen fewer per molecule, Fig. 3-2*a*.



Hexadien (1,5), C_6H_{10}

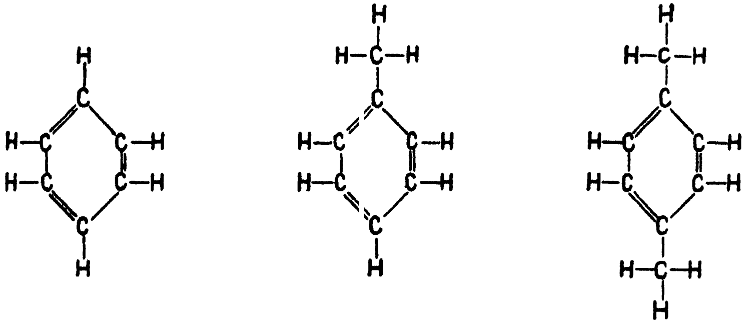
FIG. 3-3.—A diolefin (C_nH_{2n-2}).

The open-chain hydrocarbons, paraffins and olefins, are classed jointly as *aliphatics*.

The *naphthenes* have the same chemical formula, C_nH_{2n}, but are *closed-chain* hydrocarbons, Fig. 3-2*b*.

The *diolefins*, C_nH_{2n-2}, are unsaturated hydrocarbons with two double-bonded molecules, Fig. 3-3. Acetylene is one of this group. Liquid diolefins are present only in small quantities in commercial fuels.

The *aromatic* series of hydrocarbons, C_nH_{2n-6}, have a *ring-type* structure for all or most of the carbon atoms, to which are attached H or groups of C and H atoms. Three hydrocarbons with the typical benzene ring have structures as shown in Fig. 3-4.



(a) Benzene, C_6H_6 (b) Toluene, C_7H_8 (c) Xylene (para), C_8H_{10}

FIG. 3-4.—Aromatic hydrocarbons (C_nH_{2n-6}).

The paraffins and the naphthenes are saturated compounds, meaning that all the carbon atoms are fully engaged and each is attached to four other atoms. Such compounds are chemically stable. The olefins and diolefins are unsaturated compounds and therefore unstable; the carbon atoms with a double bond can attach themselves to and absorb additional atoms of hydrogen, oxygen, or sulphur if such are present.

At atmospheric conditions, hydrocarbon molecules with a low number of carbon atoms, 1 to 4, are gases. Hydrocarbons with 5 to 15 carbon atoms are more or less volatile light oils, and those with 16 to 26 carbon atoms are referred to as heavy oils. Commercial fuels and lubricating oils are mixtures of many kinds of hydrocarbons in various proportions.

Another group of hydrocarbons consists of methyl, CH_4O, ethyl, C_2H_6O, and butyl, C_4H_9OH, alcohols. These are not true hydrocarbons, since each contains oxygen in the molecule.

3-2. Gaseous Fuels.—*Natural gas* is found in many parts of the United States. By extensive pipe lines it is made available many hundreds and in some cases even thousands of miles away from the wells from which it is obtained.

Natural gas obtained from oil wells is called *casing-head gas* and is usually treated for the recovery of gasoline, after which, called *dry gas*, it is delivered into the pipe-line systems to be used as fuel.

The analysis of natural gas varies considerably with the location. The limiting values for each component can be seen from Table 3-1, which also gives specific gravity and heat values of different natural gases found in the United States. An additional column gives data referred to an imaginary average gas.

Coke-oven gas is obtained as a by-product when making coke, and its analysis depends upon the coal used and also upon the method of operating the oven. Table 3-1 shows that its heat value per cubic foot is only about one-half that of natural gas, but it requires about half the air for combustion, and the heat value of the actual air-gas mixture is practically the same as when natural gas is used.

Blast-furnace gas is a by-product of melting iron ore. Its analysis varies considerably with the fuel used and the method of operating the blast furnace. Actual and average data given in Table 3-1 show that the heat value of the gas-air mixture is only slightly less than that of the first two fuels.

Part of the gas is used for preheating the air necessary to operate the blast furnace itself and the blowing engines. The rest, about one-fourth of the total amount, can be used as engine fuel for power production. For use in gas engines the blast-furnace gas must be thoroughly cleaned, for it contains from 1 to 1½ grains of dust per cubic foot. This dust acts as a grinding

TABLE 3-1.—DATA ON COMPOUND GASEOUS FUELS

Composition, per cent by volume	Natural gas		Coke-oven gas		Blast-furnace gas		Producer gas		Sewage-sludge gas	
	Limits	Aver.	Limits	Aver.	Limits	Aver.	Limits	Aver.	Limits	Aver.
Hydrogen, H_2	0–26	......	48–57	50	2–5	3.5	10–20	15	1–5	2
Methane, CH_4	32–98	83	27–36	31	0.2–1.6	0.5	0–4	2	60–77	67
Ethylene, C_2H_4	......	......	2–4	3	......	......	0–0.5			
Ethane, C_2H_6	0–20	15								
Other hydrocarbons	0–2	......	0–1	......	......	......	0–1			
Carbon monoxide, CO	0–3	......	3–6	5	25–28	27	20–29	24	0–1	
Carbon dioxide, CO_2	0–26	1	1–3	2	6–13	11	4–8	5	14–30	25
Oxygen, O_2	0–2	......	0–12	1	0–2	......	0.2–2	1	0–1	0.5
Nitrogen, N_2	0–20	1	2–13	8	56–63	58	50–58	53	2–14	5
Hydrogen sulphide, H_2S	0–2	......	......	......	......	......	......	......	0–2	0.5
Specific density (air = 1)	0.57–0.89	0.62	0.35–0.42	0.40	0.80–1.02	1.00	0.80–0.90	0.84	......	0.81
Heat value, lower, Btu/cu ft	760–1350	980	400–500	475	90–110	100	110–170	135	600–760	625
Theoretical air, cu ft/cu ft	......	10.5	......	4.7	......	0.78	......	1.12	......	7.06
Actual air, cu ft/cu ft	......	13.5	......	6.0	......	0.94	......	1.4	......	8.9
Excess air, per cent	20–40	28	......	28	......	21	......	25	......	26
Heat value of mixture, Btu/cu ft	......	68	......	68	......	52	......	56	......	67

compound on the cylinders and pistons. The above-mentioned dust content may be better appreciated if it is stated that the gas necessary to operate a 1000-bhp engine yields from 14 to 21 lb of dust per hour.

The cleaning of the gas is usually done in two or three steps. Part of the dust is removed by allowing it to settle out in stationary dry-dust catchers. In the second stage wet scrubbers and wet revolving washing machines are used, and after that the gas sometimes is led through sawdust scrubbers. Proper cleaning can reduce the dust content to about 0.0054 gr/cu ft. A dust content of 0.01 gr/cu ft is usually considered satisfactory, meaning a cleaning efficiency of 99 to 99.5 per cent. This leaves, in the case of a 1000-hp engine running 24 hr a day, from 600 to 1200 lb of dust carried into the cylinders per year.

Producer gas as fuel for gas engines is not used in the United States much. The causes of this are the availability of natural gas and improvements in mechanical-injection oil engines which are simpler and more economical in operation than a gas-producer and engine unit.

Lately, however, interest has been shown in use of producer gas for motor transport.[1] Coal and charcoal are used as fuel. Because of absence of detonation, high compression ratios, 8:1 and up to 15:1, can be used without trouble. A relatively low heat value of the gas, 120 to 133 Btu/cu ft, and a relatively high temperature of the gas coming from the producer, which results in a low charge efficiency of the engine, reduce the power developed to only about 60 to 65 per cent of that obtained with gasoline. However, owing to a higher compression ratio, the thermal efficiency is about the same as with gasoline but drops faster with a decrease of the air-fuel ratio.

Producer-gas analysis varies with the fuel used, anthracite, coke, bituminous coal, or wood, and dampness of air, which influences the hydrogen content, as shown in Table 3-1.

Sewage-sludge Gas.—With the development of sewage-disposal plants, sewage-sludge gas is produced and used as fuel for internal-combustion engines.[2] These engines furnish energy for driving the pumps in the sewage plants. The composition of the gas varies considerably. A typical analysis for gas from metropolitan wastes is given in Table 3-1.

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[1] ALLCUT, E. A., Producer Gas for Motor Transport, *Automotive and Aviation Ind.*, vol. 89, No. 2, p. 38, Aug. 1, 1943, and No. 3, p. 39, Aug. 15, 1943.

[2] WALRAVEN, W. B., Adapting Gas Engines to Sewage-sludge Gas, *Mech. Eng.*, vol. 59, p. 835, November, 1937.

3-3. Gasoline.—The characteristics of gasoline, chiefly the temperatures of distillation, often, although wrongly called volatility, vary considerably depending upon the nature of oil or gas from which gasoline is obtained and the methods of preparation.

Four main methods are applied for making gasoline:

1. Distillation or refining from crude oil.
2. Cracking residue oil.
3. Polymerization of gases produced by the cracking process, which produces a high-octane gasoline that can be used either directly or blended with gasoline produced by other methods.
4. Extraction from natural gas by absorption and distillation and blending with some heavier hydrocarbons, for the gasoline from gas is so volatile that it cannot be used alone.

A fifth method—extraction from natural gas by compression and cooling—now is used very little, having been superseded by the absorption method.

Finally a sixth method should be mentioned—obtaining gasoline from hydrogenized coal. At present, synthetic gasoline is produced on a commercial scale in Germany and Great Britain. However, the production cost of synthetic gasoline is considerably higher than that of gasoline obtained from petroleum. The characteristics of synthetic gasolines in general do not differ materially from the characteristics of petroleum gasoline.

Commercial gasoline obtained by any one of the methods is a mixture of the three main series of more or less volatile hydrocarbons and contains from 10 to 80 per cent paraffins, C_nH_{2n+2}, from 15 to 85 per cent naphthenes, C_nH_{2n}, and from 4 to 40 per cent aromatics, C_nH_{2n-6}. However, to simplify chemical calculations, octane, C_8H_{18}, is often used as a fair average of commercial gasoline except, of course, for distillation temperatures. Its molecular weight, 114, and hydrogen content, 15.8 per cent, are very close to the average values for commercial gasolines, 113 and 15.5, respectively.

Distillation Test.—Gasoline is supposed to contain fractions which boil at atmospheric pressure between 120 and 300 F. However, actual testing shows that gasoline usually contains fractions of higher temperatures. Plotting amounts distilled according to the ASTM specifications[1] against temperatures of distillation gives the so-called ASTM *distillation curves*, Fig. 3-5, which show clearly the variation in the characteristic of different gasolines. Curves *A* and *E* represent commercial gasolines obtained by cracking; *B* is a special-blended gasoline of a very uniform composition; however, the absence of low-temperature fractions makes the starting of a cold engine hard. *C* is a commercial low-test gasoline. Curves *F*, *D*, and *M* represent characteristics of gasolines as required by United States government specifications for Fighting (*F*) and Domestic (*D*) grades of aviation gasolines and Motor (*M*) gasoline.

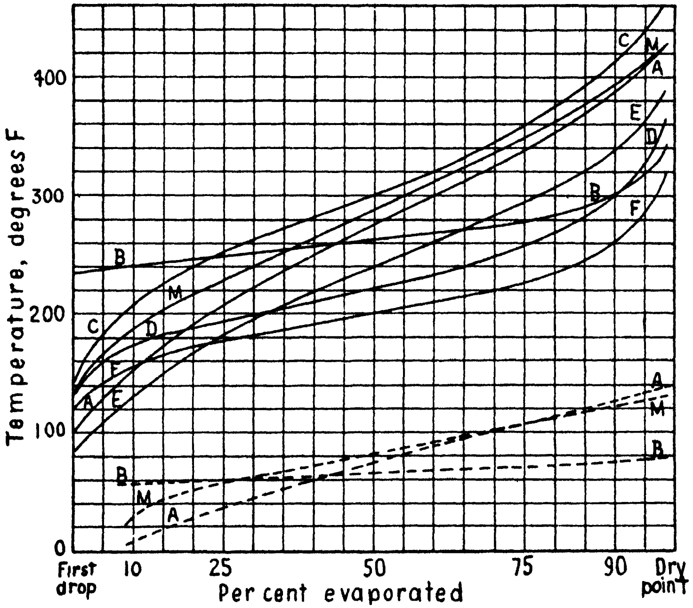

Fig. 3-5.—Distillation and volatility curves of gasolines.

In the Engler distillation test the flask contains gasoline and gasoline vapor. so that the distillation curves give temperatures of saturated vapor at atmospheric pressure. In an internal-combustion engine, gasoline vapors are burned mixed with air, and, although the pressure of the mixture during the suction stroke may be approximately atmospheric, the pressure of gasoline vapor is only part of the total pressure. According to

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[1] ASTM Standard No. 86.

Dalton's law the temperature of evaporation corresponds not to the total pressure of a gas-vapor mixture but to the partial pressure of the vapor. Therefore the distillation curves from an Engler test, Fig. 3-5, cannot be used directly for any problems of vaporization as encountered in engine carburetors. These curves give information only as to comparative value of different grades of volatile fuel.

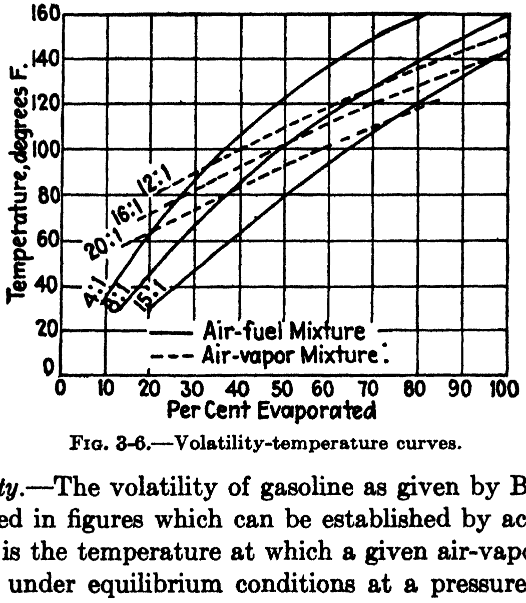

FIG. 3-6.—Volatility-temperature curves.

Volatility.—The volatility of gasoline as given by Bridgeman[1] is expressed in figures which can be established by actual tests: volatility is the temperature at which a given air-vapor mixture is formed under equilibrium conditions at a pressure of 1 atm (14.7 psia) when a given part of the fuel is evaporated.

Volatility thus defined may be best explained by an example. In Fig. 3-6 curves are plotted in full lines, showing the relation between amounts of fuel evaporated and the temperature to which the mixture had to be raised, found from actual tests using different air-fuel mixtures. As the proportion of air to fuel by weight is decreased, the temperatures increase because the partial pressures of the vapor increase.

From the air-fuel curves, points of the air-vapor curves shown in dotted lines were determined. Evidently an air-vapor ratio r_{av} is equal to the air-fuel ratio r_{af} divided by the part evaporated x_e:

$$r_{av} = r_{af}/x_e \tag{3-1}$$

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[1] *SAE Journal*, vol. 22, No. 4, p. 437, 1928.

The intersection of an air-vapor curve with a vertical line drawn through the amount evaporated gives the volatility for the conditions as defined before. The air-vapor curves as shown in Fig. 3-6 are therefore called *volatility curves.*

Such volatility curves for a 16:1 air-vapor ratio are shown in the lower part of Fig. 3-5 for some of the gasolines whose distillation curves are shown in the same figure.

Example 3-1.—Find the temperature required to produce a 20:1 air-vapor mixture for 40 per cent evaporated fuel for gasoline having the volatility characteristics given in Fig. 3-6.

The 20:1 curve intersects the ordinate through 40 per cent at 86 F.

Example 3-2.—What part must be evaporated to obtain a 20:1 air-vapor mixture from a 4:1 air-fuel mixture?

From expression (3-1), part evaporated,

$$4/20 = 0.2 \text{ or } 20 \text{ per cent}$$

The same answer may be obtained directly from Fig. 3-6 as the point of intersection of the 4:1 air-fuel distillation curve and the 20:1 volatility curve.

Empirical Expression.—The determination of air-fuel distillation curves or points, Fig. 3-6, necessary to plot volatility curves, is a considerably more difficult task than the determination of the ASTM distillation curves by the Engler method, Fig. 3-5.

The following empirical expression simplifies this problem by giving the relation between the ASTM distillation temperatures T_e, in degrees Rankine, for a given part evaporated x_e and the volatility temperature T_v for a resultant air-vapor mixture r_{av}:

$$T_v = T_e/[1.372 + 0.005(r_{av} - 16) \pm 0.45(x_e - 0.5)^2] \quad (3\text{-}2)$$

where the + sign is used for $x_e > 0.5$ and the − sign for $x_e < 0.5$.

This formula can be used for r_{av} from 8 to 20 and for x_e from 0.05 to 0.90.

Example 3-3.—Find the lowest temperature at which an engine can be started using fuel *A*, Fig. 3-5, if a 1:1 mixture of air to fuel is the richest mixture the carburetor can furnish.

Knowing that the leanest air-vapor ratio which will ignite $r_{av} = 20$, the part evaporated for $r_{af} = 1$ from expression (3-1) is

$$x_e = r_{af}/r_{av} = 1/20 = 0.05$$

From Fig. 3-5 for $x_e = 0.05$ the temperature is 129 F or 589 R; from expression (3-2),

$$T_v = 589/[1.372 + 0.005(20 - 16) - 0.45(0.05 - 0.5)^2] = 452 \text{ R}$$

or

$$t_v = 452 - 460 = -8 \text{ F}$$

Dew Point.—The dew point of an air-gasoline-vapor mixture is the temperature of saturation of vapor at a given pressure. The dew point gives the lowest temperature at which a given air-vapor mixture can exist without condensation.

The following empirical formula[1] gives comparatively accurate values for the dew point T_{dp}, degrees Rankine, expressed as a function of the distillation temperature T_{90}, degrees Rankine, found by the ASTM method, for 90 per cent evaporated:

$$T_{dp} = T_{90}/\{1.084 + 0.185 \log_{10} [(1 + 3.9r_{av})(14.7/p)]\} \qquad (3\text{-}3)$$

where p is the pressure in pounds per square inch absolute of the mixture.

Specific gravity is the ratio of the weight of a certain volume of a substance to the weight of an equal volume of distilled water at the same temperature. Specific gravity is a certain characteristic of a liquid fuel.

Another scale of density is the arbitrary Baumé scale. For liquids lighter than water the following relation exists between degrees of the Baumé scale and specific gravity γ at 60 F:

$$°\text{Bé} = 140/\gamma - 130 \qquad (3\text{-}4)$$

The American Petroleum Institute uses the Baumé scale with slightly changed constants:

$$°\text{API} = 141.5/\gamma - 131.5 \qquad (3\text{-}5)$$

Specific gravity of gasoline used as engine fuel varies from about 0.71 to 0.76, or 67° to 54°Bé, or 67.8 to 54.7°API.

The specific gravity of various hydrocarbons, except that of the aromatics, increases with their molecular weight, as shown by the curves of Fig. 3-7.[2]

Commercial gasolines are mixtures of different hydrocarbons, and there is no definite relation between their gravity and volatility. However, the volatility usually is greater with a lower specific gravity, or greater number of degrees Baumé or API.

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[1] BRIDGEMAN, O., Dew Point Data on Gasoline, *SAE Journal*, vol. 23, No. 5, p. 484, 1928.

[2] SPAUSTA, FRANZ, "Treibstoffe für Verbrennungsmotoren," p. 233, Julius Springer, Vienna, 1939.

Specific heat of liquid gasoline and of gasoline vapor can be determined from one of several empirical formulas, expressing

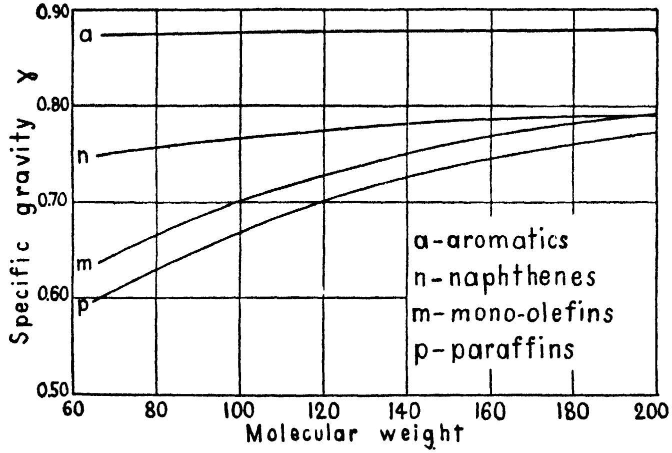

FIG. 3-7.—Influence of structure and molecular weight on specific gravity of hydrocarbons.

c as a function of temperature and specific gravity. For liquid gasoline the following simple formula may be used:

$$c = (2.1 - \gamma)(t + 670)/2030 \text{ Btu/lb} \qquad (3\text{-}6)$$

Specific heat of gasoline vapor at atmospheric pressure can be determined from a similar formula

$$c_p = (4 - \gamma)(t + 670)/6450 \text{ Btu/lb} \qquad (3\text{-}7)$$

Both formulas may be used for any hydrocarbon with a specific gravity γ between 0.68 and 0.90.

3-4. Performance in Engines. *Temperature Drop.*—The latent heat of evaporation of gasoline absorbs heat from the air and lowers its temperature. In an engine the evaporation of gasoline occurs partly in the carburetor, partly in the intake manifold, and often is completed only in the cylinder.

The drop in temperature of the air charge does not depend upon where evaporation occurs, provided that it is completed before the beginning of the compression stroke.

This drop in temperature t_d can be computed using the same designations as before from equation

$$x_e L = r_{af} c_p t_d \tag{3-8}$$

where L is the latent heat of vaporization and $c_p = 0.24$ is the specific heat of air. L can be taken from Fig. 3-8,[1] which gives the relations between temperature, pressure, latent heat, and specific volume for different hydrocarbons. The pressure is the partial pressure p_f and can be computed. Using expression (2-13), which gives $p_f = pr_{vf}$, substituting for r_{vf} by expression (2-20) $r_{vf} = (r_{wf}/M_f)/(r_{wf}/M_f + r_{wa}/M_a)$, simplifying the results, and noticing that in the carburetor $r_{wa}/r_{wf} = r_{av}$, this gives

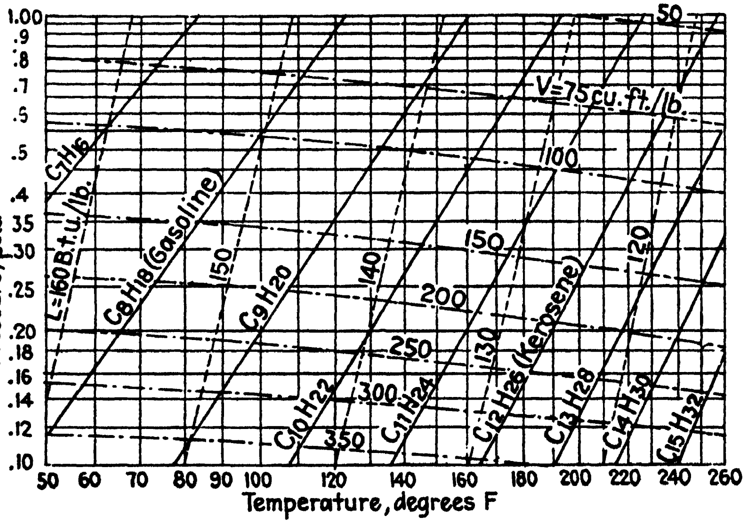

FIG. 3-8.—Latent heat of vaporization and specific curves for hydrocarbons as functions of pressure and temperature.

$$p_f = p/(1 + r_{av} M_f/M_a) \tag{3-9}$$

where p is the total pressure of the mixture, M_f and M_a are the molecular weights of the fuel and air, respectively, and $M_a = 29$. Commercial gasoline being a blend of a great number of various hydrocarbons, its temperature-pressure curve, Fig. 3-10, runs considerably higher than that of octane.

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[1] EDMISTER, W. C., and BUSKELL, C., Thermodynamics of Paraffin Hydrocarbons, *Oil Gas J.*, July 2, 1931, p. 22.

Heating of Charge.—The drop in temperature from equation (3-8) is added to the equilibrium temperature of the entering air necessary to obtain a certain air-vapor ratio r_{av}. If the air temperature is lower than $t_v + t_d$, the charge must be heated.

Gasoline of an average grade can give at 68 F air temperature only about 40 per cent evaporation.

The heat Q which must be supplied to obtain a certain air-vapor mixture consists of three parts:

1. Heat to raise the air temperature to the equilibrium temperature.
2. Heat to raise the fuel temperature to the same temperature.
3. Latent heat of evaporation of the required part of fuel.

Combining the two first amounts gives for 1 lb gasoline

$$Q = (r_{af}c_p + c_f)(t_v - t_1) + Lx_e \tag{3-10}$$

where the specific heat c_f of liquid gasoline is found by expression (3-6).

Heat necessary for complete vaporization is found from equation (3-10) with $x_e = 1$. However, in an actual engine the amount of heat which must be furnished is less than Q because heat is conducted and radiated to the carburetor and intake manifold from the hot exhaust manifold and from the hot air from the radiator, and also because part of the fuel is vaporized only after it has entered the cylinder.

***Example* 3-4.—Determine the heat which must be supplied per pound of gasoline to make possible 50 per cent vaporization in the intake manifold when the outside temperature is 50 F, fuel used is gasoline C in Fig. 3-5, and the vacuum in the manifold is 1.4 in. Hg; the carburetor gives $r_{af} = 10:1$; $\gamma = 0.74$.**

From expression (3-1),

$$r_{av} = 10 \div 0.5 = 20$$

The distillation temperature t_e for 50 per cent is 302 F, Fig. 3-5, and from equation (3-2),

$$T_v = 762/[1.372 + 0.005(20 - 16) + 0.45(0.5 - 0.5)^2] = 548 \text{ R, or } 88 \text{ F}$$

The partial pressure of saturated gasoline vapor, C_8H_{18}, with

$$m_f = 8 \times 12 + 18 = 114$$

and $p = 14.7 - 1.4 \times 0.491 = 14.01$, is from expression (3-9)

$$p_f = 14.01/(1 + 20 \times 114/29) = 0.176$$

From Fig. 3-8 for this pressure $L = 157$; by expression (3-6)

$$c_p = (2.1 - 0.74)(88 + 670)/2030 = 0.51$$

thus from expression (3-10),

$$Q = (10 \times 0.24 + 0.51)(88 - 50) + 157 \times 0.5 = 110.6 + 78.5 = 189.1 \text{ Btu/lb}$$

Example 3-5.—Determine the heat which must be supplied for the conditions of Example 3-4 but for complete vaporization and $r_{af} = 15:1$.

With t_{90} for 90 per cent being 414 F, Fig. 3-5, the dew point t_{dp} is found from equation (3-3), for $r_{av} = 15:1.0 = 15.0$

$$T_{dp} = 874/\{1.084 + 0.185 \log [(1 + 3.9 \times 15.0)(14.7/14.01)]\} = 617 \text{ R, or } 157 \text{ F}$$

by expression (3-6)

$$c_p = (2.1 - 0.74)(157 + 670)/2030 = 0.56$$

Thus, from expression (3-10),

$$Q = (15 \times 0.24 + 0.56)(157 - 50) + 157 = 455 + 157 = 612 \text{ Btu/lb}$$

These examples show that the heat necessary for a given evaporation depends upon the volatility characteristic of the fuel, the air-fuel ratio, the outside temperature, the pressure in the manifold, and the latent heat of the fuel.

Detonation.—A very sudden rise in pressure during combustion accompanied by a metallic, hammerlike sound is called *detonation*, or knocking. Detonation occurs near the end of the pressure rise after ignition of the fuel charge. Detonation is a function of many factors, the main ones being the chemical composition and molecular structure of the liquid fuel. Some fuels detonate more easily, some less, and some, such as benzol or ethyl alcohol, do not detonate. Ordinary gasoline is made less likely to detonate by adding small amounts of special compounds, such as tetraethyl lead, $(C_2H_5)_4Pb$, or ethyl iodide, C_2H_5I. One part by volume of tetraethyl lead in about 1300 parts of gasoline has a very definite detonation-suppressing effect; the same effect is obtained by one part of ethyl iodide in about 25 parts of gasoline or 1 part of ethyl alcohol in about 8 parts of gasoline.

Octane Number.—The development of antiknock gasoline is based on a method for measuring the knocking property of a gasoline and expressing it by a figure. The method consists in testing the gasoline in question in a special C.F.R.[1] test engine

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[1] DICKINSON, H. C., The Cooperative Fuel Research and Its Results, *SAE Journal*, vol. 25, No. 2, p. 171, 1929.

TABLE 3-2.—GASOLINE SPECIFICATIONS

User	U. S. Army			U. S. Navy		Domestic Commercial Airlines					Fed. Specif. Board	
Service	Aviation			Aviation		Aviation					Passenger cars, trucks	Fire engines, ambulances
Specification	2-93-A	2-95-A	2-92-A	M-222-a	M-302-a	ASTM-73	ASTM-80	ASTM-87	ASTM 90	ASTM 95	VV-G-101-a	VV-M-571-a
Date introduced	4-15-38	6-19-35	6-20-35	9-15-34	2-1-36	....	....	....	....	....	4-13-36	4-16-36
Grade designation	65	92	100	Low oct. base	High oct. base	73	80	87	90	95	U. S. Motor gasoline	Motor fuel V
Octane number	65	92	100	73, 87	87,[a] 87, 100	73	80	87	90	95		65-75
Test engine	C.F.R.[b]	C.F.R.[b]	C.F.R.[b]	C.F.R.[c]	C.F.R.[c]	C.F.R.	C.F.R.	C.F.R.	C.F.R.	C.F.R.	....	C.F.R.[c]
Max. tetraethyl lead cc[e] per gal	0	6.0	3.0	0, 3, 27	0, 0.5, 3.0	0	2.0	3.0	4.0	4.0	....	3.0
Corrosion	[d]	[d]	[d]	[d]	[d]	no	no	no	no	no	[e]	[e]
Gum, mg/100 cc[c]	.....	6[f]	6[f]	6[f]	6[f]	6,[f] 5[g]	6,[f] 5[g]	6,[f] 5[g]	6,[f] 5[g]	6,[f] 5[g]		
Sulphur, max. %	0.10	0.10	0.10	0.10	0.10	0.05	0.05	0.05	0.05	0.05	0.10	0.10
Distillation, min.:												
10% evap. at °F	167	167	167	167	167	158	158	158	158	158	167	149[h]
50% evap. at °F	230	212	212	212	212	212	212	212	212	212	248	257[i]
90% evap. at °F	320	275	275	275	275	257	257	257	257	257	392	356[j]
10 + 50% point °F	307	307	307	307	307	307	307	307	307	307		
End point, max. °F	.....	.....	.....	......	......	311	311	311	311	311		
Recovery, min. %	.....	.....	.....	......	......	97	97	97	97	97		
Residue, max. %	2	2	2	2	2	1.5[k]	1.5[k]	1.5[k]	1.5[k]	1.5[k]	2	2
Loss, max. %	.....	.....	.....	......	......	1.5	1.5	1.5	1.5	1.5		
Reid vapor press. at 100°F., max. psi	7	7	7	7	7	7	7	7	7	7	12	10[l]
Freezing point, °F	−76	−76	−76	−76	−76							
Other requirements	[m]	[m]	[m]	[m]	[m]	[n]	[n]	[n]	[n]	[n]		

[a] This unleaded grade is used for postoverhaul testing of high-output engines and for operation prior to placing engines in storage.

[b] Special C.F.R. engine cylinder, not plug method; U. S. Army specification 2-94 (7-12-35).

[c] ASTM test method D-357-387 standard.

[d] 100 cc of the fuel evaporated to dryness in a weighed polished copper dish shall cause no gray or black corrosion, and the increase in the weight of the dish shall not exceed 5 mg.

[e] ASTM test method D 130-30.

[f] The fuel shall be subjected to an accelerated aging test at 212 F with oxygen at approximately 100 psi pressure, and in the presence of iron for 5 hr. On evaporation of 100 cc of a mixture of oxidized sample and gum solvent, the amount of residue shall not exceed 6 mg.

[g] Gum test conducted in a polished copper dish.

[h], [i], and [j] These temperatures are raised 14, 17, and 19 F, respectively, for an altitude of 8000 ft above sea level.

[k] Acidity of residue must be neutral.

[l] The Reid vapor-pressure limit is lowered 1.5 psi for each 5000 ft elevation above sea level.

[m] 80 cc of the fuel shall be shaken in a glass-stoppered graduate with 20 cc distilled water at room temperature. On settling after making, the volume of the aqueous layer shall not have increased or decreased by more than 2 cc.

[n] Doctor test must be negative; color shall be blue.

with variable compression and in comparing it with a certain mixture whose knocking tendency is known. It has been established that pure iso-octane, C_8H_{18}, is practically free from knock, whereas pure heptane, C_7H_{16}, knocks worse than any known gasoline. Mixtures of iso-octane and heptane can be made to duplicate or match the knocking property of any gasoline. Such a mixture is designated by the percentage of octane, and the term *octane number* has resulted. Thus the octane number of a gasoline is the percentage of octane in a mixture with heptane which, when tested in the test engine, will knock to the same extent as the gasoline in question. This characteristic is expressed in whole numbers, as 68, or 72, etc. The greater the octane number, the better is the antiknocking property of the gasoline.

By blending gasoline with isopropyl ether, $C_6H_{14}O$, modifying the test procedure, and applying extrapolation, fuels are obtained, chiefly for aviation use, with an octane rating above 100 and up to 125 o.n.[1]

Table 3-2 gives specifications of gasolines for various types of service.

3-5. Fuel Oils.—Oils used as fuel in oil engines differ in their composition and characteristics, or specifications, still more than gasolines. Commercial fuel oils generally are residues left after the more volatile fractions, gasoline and kerosene, have been removed.

Air-injection and low-speed airless-injection engines are using a wide variety of liquid fuels, from a light grade, $\gamma = 0.82$, down to the heaviest ones, $\gamma = 0.95$. High-speed engines require a light oil, $\gamma = 0.82$ to about 0.87. These oils contain 12.5 to 13.5 per cent of hydrogen and have an apparent molecular weight of 215 to 225. These fuel oils are blended from different runs. For many calculations it is helpful to use a single hydrocarbon such as octane, C_8H_{18}, is used for gasoline. Dodecane, $C_{12}H_{26}$, was used before in such calculations; however, a better substitute is hexadecadiene, $C_{16}H_{30}$, with a hydrogen content of 13.6 per cent and a molecular weight of 220.[2]

Table 3-3 gives average compositions for fuels oils of different origin.

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[1] *J. Inst. Petroleum Tech.*, vol. 27, No. 211, p. 191, May, 1941.

[2] TAYLOR, C. F., and TAYLOR, E. S., "The Internal Combustion Engine," p. 16, International Textbook Company, Scranton, Pa., 1938.

Curve *D* in Fig. 3-9 shows that distillation temperatures of a diesel fuel oil may be close to those of kerosene, curves *A* and *B*,

TABLE 3-3.—COMPOSITION OF FUEL OIL FOR COMPRESSION-IGNITION ENGINES,[a] PER CENT

Origin	Paraffins C_nH_{2n+2}	Olefins C_nH_{2n}	Naphthenes C_nH_{2n}	Aromatics C_nH_{2n-y}
California	40–65	2–18	14–20	10–30
Mid-continent	50–74	2–14	15–24	3–16
Pennsylvania	70–75	2– 5	16–19	4– 9

[a] JOACHIM, W. F., and NUTT, H. V., Fuels for Diesel Engines in Marine Transportation, *Trans. ASME*, vol. 62, No. 7, p. 597, October, 1940.

or may run higher, curve *E*, which refers to a fuel oil for high-speed engines.

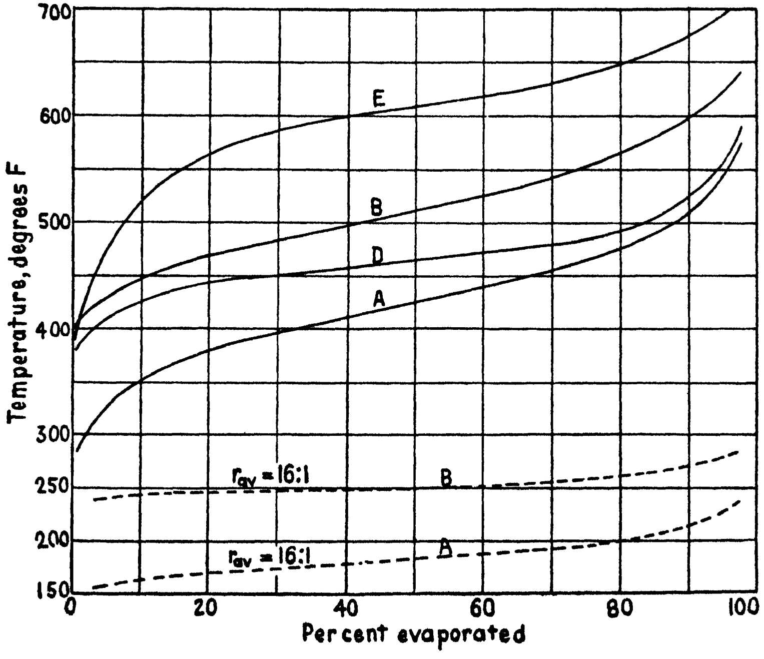

FIG. 3-9.—Distillation and volatility curves of kerosenes and diesel oils.

Specifications.—There are no generally accepted specifications. Many quantity purchasers of fuel oil, as the United States government and several oil-engine manufacturers, write their own specifications. Specifications differ depending upon the type of engine; low-speed engines are able to use a heavier oil which would not be satisfactory in high-speed engines.

The ASTM has proposed some tentative specifications with the object of having a more universal standard. All these various specifications cover more or less the same items. The ASTM has established detailed specifications for carrying out the laboratory tests in respect to each of the fuel specification items.[1]

Viscosity is an important characteristic, as it affects the atomization of fuel and operation of the high-pressure fuel pumps.

The greater the number of carbon atoms, the greater the viscosity of the oil. Of two hydrocarbons having the same number of carbon atoms, the one with the lower hydrogen content will have a higher viscosity and its viscosity will change more rapidly with the change of temperature.

In American practice, viscosity is determined by the Saybolt Universal viscosimeter. Light distillate engine fuel oils have a viscosity of from 35 to 175 sec at 100 F. Intermediate fuels may have a viscosity up to 500 sec Saybolt Universal, designated 500 SSU; viscous oils may have a viscosity up to 100 sec Saybolt Furol, and very viscous special fuels up to 300 sec Furol, measured at 122 F. The Saybolt Furol scale readings are converted to the Universal Saybolt scale by multiplying them by 15. A viscosity above 100 SSU is not desirable, as it requires heating of the oil in order to lower the viscosity and make it flow. A viscosity lower than 35 SSU is dangerous, as the oil will have a very low lubricating value and may cause excessive wear and even seizing of the fuel-pump plungers; it will also produce an excessive leakage past the plungers.

For air-injection engines the viscosity limit can be raised up to 100 sec and even 300 sec Furol at 122 F.

Flash Point.—A minimum flash point of 150 F is specified for safety. For some high-speed engines, specifications lower this value down to 115 F.

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[1] ASTM 1941 Book of Standards. A number of specifications for various engines are given in a review by W. F. Joachim and H. V. Nutt, Fuels for Diesel Engines in Marine Transportation, *Trans. ASME*, October, 1940, pp. 595–604. Hubner, W. H., Murphy, A. B., and Egloff, G., A Study of Diesel Fuel Specifications, *Universal Oil Products Booklet* 228, revised 1939.

Water and sediment, determined according to ASTM, should not exceed 0.5 per cent.

Sulphur content should not exceed 2 per cent and if possible not 0.7 per cent, because sulphur dioxide in the presence of water vapor has a highly corrosive action on steel and even cast iron, especially at high temperatures.[1]

Specific gravity or equivalent degrees API have no direct bearing upon the burning qualities of an oil. However, the above given limitations of viscosity more or less confine the limits of specific gravity to about 0.83 to 0.90, or 40 to 24°API, for airless-injection engines and down to 0.94, or 19°API, for air-injection engines.

Ash content should not exceed 0.12 per cent by weight for the heaviest fuel and should be 0.01 per cent for light fuels used in high-speed engines.

Corrosiveness.—Fuel oil must not contain any free acids.

Ignition Quality.—The ignition quality of a fuel is one of the most important characteristics; it is a measure of its ability to ignite promptly after injection, thus ensuring a progressive smooth burning and easy starting.

In a compression-ignition engine the ideal fuel should ignite immediately after being injected. Actually there is always a certain lapse of time between the beginning of injection and ignition of the fuel. This time element is termed *ignition lag* or *delay*. If the ignition lag is large, the whole fuel charge may be injected before it ignites; and when it ignites, the finely atomized charge burns suddenly with a great pressure rise and often produces a sharp sound similar to the knocking sound of a spark-ignition engine. Thus the *combustion knock* in a compression-ignition engine, contrary to a spark-ignition engine, is caused by a fuel which ignites too slowly. In larger, low-speed engines the ignition lag is not so noticeable and the usual fuels give a satisfactory performance. In high-speed engines, with small cylinders and high rotative speeds, the time allowed for combustion is very short and a small ignition lag, meaning good ignition quality, becomes very important.

Evidently, a fuel with a lower ignition temperature, all other conditions being equal, will ignite more quickly when injected into the combustion chambers than one with a higher ignition temperature. However, the ignition quality of a fuel depends also on other factors. The best way to determine the quality is to test the fuel in the engine.

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[1] *Op. cit., Trans. ASME*, October, 1940, p. 600.

So far there is no standard method for determining and measuring the ignition quality. Many different methods have been proposed: (*a*) the cetane number, which is the most used method, (*b*) the critical compression ratio, (*c*) the diesel index, and (*d*) the aniline point.[1]

The *cetane number* is similar to the octane number: The performance of the fuel in question is compared in a C.F.R. test engine with that of a reference fuel.[2] The original primary reference fuel used for this method is cetane, a saturated polyolefin, $C_{16}H_{34}$, which is mixed with α-methylnaphthalene, $C_{11}H_{10}$, a very slowly igniting fuel. The percentage of cetane in the mixture which has the same ignition quality as the fuel which is tested is called the *cetane number.* Thus a fuel with a cetane number 45 has the same ignition quality as a mixture of 45 per cent $C_{16}H_{34}$ and 55 per cent of $C_{11}H_{10}$. These hydrocarbons are used because compression-ignition engines run very smoothly on $C_{16}H_{34}$ and knock very badly on $C_{11}H_{10}$. The recommended minimum cetane numbers are 25 to 35 for low-speed, 35 to 45 for medium-speed, and 45 to 50 for high-speed engines.

The determination of the cetane number is made by two different methods: by the critical compression ratio, C.C.R., and by the delay method.

The primary reference fuel, $C_{16}H_{34}$, is expensive and not very stable. In practice secondary reference fuels are used: *e.g.*, a Shell secondary diesel reference fuel as high-reference fuel and commercial methylnaphthalene as low-reference fuel.

Cetene Number.—Originally the reference fuel used was cetene, $C_{16}H_{32}$. However, cetene is more difficult to produce and is rather unstable and also is less ignitable than cetane, a cetane number of 76 being approximately equal to a cetene number of 100. For these reasons cetene numbers are gradually disappearing from use.

Cetane-octane Relation.—There seems to exist a simple, linear relation between the characteristic numbers, octane (o.n.) and cetane (c.n.). One of these formulas, supposed to be accurate within ±5 per cent is[1]

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[1] HETSEL, T. R., The Development of Diesel-fuel Testing, *Penn. State Coll. Eng. Exp. Sta. Bull.* 45, 1936.

[2] *Proc. ASTM*, vol. 38, Part 1, p. 392, 1938.

$$\text{o.n.} = 120 - 2\ \text{c.n.} \qquad (3\text{-}11)$$

another one gives the relation as[2]

$$\text{o.n.} = 150 - 2.9\ \text{c.n.} \qquad (3\text{-}12)$$

Critical compression ratio, C.C.R.,[3] is the minimum compression ratio at which ignition of the fuel in a modified C.F.R. test engine with a variable compression space is obtained. The C.C.R. number is sometimes used as an ignition-quality index itself; the lower the C.C.R., the better the ignition quality of the fuel. For the same three engine groups for which the cetane numbers are given above, the minimum C.C.R. numbers are 9.8, 8.8, and 8.1.

Delay method[4] divides the combustion process into three steps:

1. The delay period between the time the first portion of fuel leaves the nozzle and the point of *appreciable pressure rise* due to burning.
2. The rapid burning of the fuel which was delivered during the delay period.
3. The burning of the rest of the fuel charge at substantially the same rate at which it is injected.

The delay period 1 is measured in degrees of the crank travel by means of a modified C.F.R. test engine with an electrical hookup which flashes when fuel injection begins and when ignition occurs, as indicated by an appreciable pressure rise.

Diesel index is computed from the expression[5]

$$\text{Diesel index} = \text{An Pt} \times {}^\circ\text{API}/100 \qquad (3\text{-}13)$$

where An Pt, or the *aniline point*, is found by heating a mixture consisting of equal volumes of the test sample and freshly distilled, water-free aniline, C_6H_7N, until a clear solution is obtained. Then, while the solution is cooling, the temperature at which turbidity appears is noted. Experiments show that the higher the diesel index, the better the ignition quality of the fuel. A simple correlation between the diesel index and the cetane number does not exist. The minimum diesel-index numbers are 20 for low-speed, 30 for medium-speed, and 45 for highspeed engines. The diesel index is not used widely, because it does not work with some fuel oils.

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[1] *Z. Ver. deut. Ing.*, vol. 84, No. 22, p. 376, 1940.

[2] *Library Bulletin of Abstracts*, Universal Oil Products Company, vol. 14, No. 36, pp. 142–143, September, 1941.

[3] POPE, A. W., JR., and MURDOCK, J. A., Compression-ignition Characteristics of Injection-engine Fuels, *SAE Journal*, vol. 30, p. 136, 1932.

[4] CHANDLER, ASME *Oil and Gas Power Proc.*, 1937.

[5] BECKER, A. E., and FISCHER, H. G. M., A Suggested Index of Diesel-fuel Performance, *SAE Journal*, vol. 35, p. 376, 1934.

Aniline Point.—Sometimes the aniline point, An Pt, as found for the diesel-index method is used as an ignition-quality index by itself. An An Pt of 140 to 150 F corresponds to a cetane number of about 35 to 45.

Other Fuel Oils.—*Bunker C oil* is a heavy, viscous residue oil used for boilers. However, when heated to about 180 F it becomes fluid enough to be used in air-injection diesel engines.[1]

Tar oil, a by-product of gas production from bituminous coal and also from coke-oven operation, is used to a great extent in Germany as fuel for diesel engines. It has a high viscosity and high temperature of ignition and a specific gravity of 0.97 to 1.11.

3-6. Other Liquid Fuels.—*Kerosene* is distilled from petroleum but is heavier than gasoline, having a specific gravity of 0.79 to 0.83 at 60 F. Figure 3-9 gives typical distillation curves of two grades of kerosene—curve *A* corresponding to "water-white" kerosene as per United States government specifications. Curve *B* is an inferior commercial grade. The two lower curves represent volatility curves for these kerosenes, both for $r_{av} = 16$.

The flash point of kerosene is not lower than 115 F.

Figure 3-9 compared with Fig. 3-5 shows that the temperatures required for a certain air-vapor mixture from a given air-fuel mixture are considerably higher for kerosene than for gasoline.

Decane, $C_{10}H_{22}$, or dodecane, $C_{12}H_{26}$, depending on the specific weight of the fuel, may be used for calculations instead of kerosene, for example, in finding the latent heat of evaporation from the chart, Fig. 3-8. The temperatures of vaporization and saturation can be computed from the same formulas (3-2) and (3-3) as in the case of gasoline, and all the other computations are identical with those for gasoline. The vapor-pressure curves of commercial kerosenes with the scale of ordinates twenty times larger than for the other fuels as indicated on the right side of the drawing are given in Fig. 3-10. They run slightly higher than the curve of dodecane, Fig. 3-8.

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[1] BARNETT, BREESKIN, Cutting Costs with Bunker C Oil, *Diesel Power and Transportation,* vol. 19, No. 2, pp. 137–40, February, 1941.

Kerosene detonates more easily than gasoline and cannot be used with a compression ratio over 4.2.

Benzol is often called *benzene* because it consists principally of benzene, C_6H_6. Commercial benzol contains about 70 per cent benzene, 24 per cent toluene, C_7H_8, and 6 per cent heavy hydrocarbons; 90 per cent of it distills off at 212 F. Its specific gravity is 0.88, specific heat 0.4, the latent heat of vaporization about 190 Btu/lb, and viscosity about 20 SSU. It is considerably less detonating than gasoline, allowing the use of a compression ratio up to 6.9. Added to gasoline, it has an antiknock effect, 20 per cent of benzol being enough to stop knocking in most gasoline engines. The vapor-pressure curve of benzene is given in Fig. 3-10.

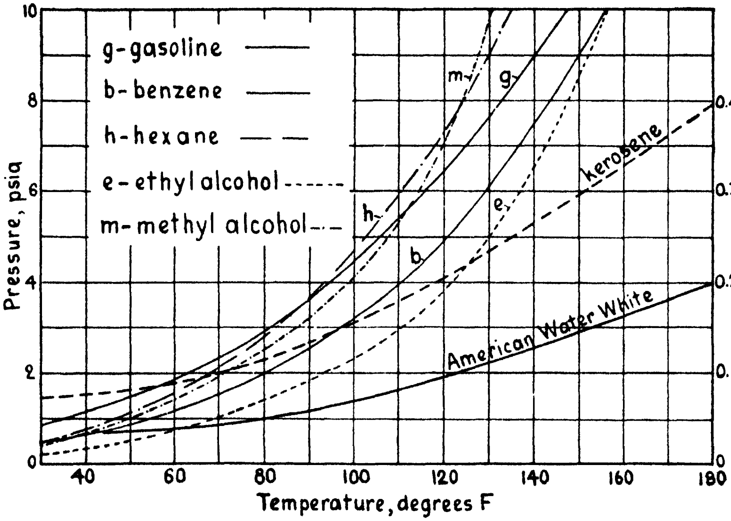

FIG. 3-10.—Vapor-pressure curves of liquid fuels.

Propane.—At a pressure of 250 psi and at ordinary temperatures, propane is a liquid and is kept in tanks or drums. It is used in truck engines and also as a stand-by fuel in gas engines.

Propane has an octane rating of 125 o.n. and therefore can be used in engines with a high compression ratio, up to 8.8.[1]

Butane.—Commercial butane, which contains about 20 per cent propane, is used instead of gasoline in both truck and stationary engines. It has a high antiknocking characteristic and permits high compression ratios up to 10.[2]

Alcohols.—*Ethyl,* or grain, alcohol, C_2H_6O, is obtained by distillation of fermented sugar solutions. Its specific gravity is 0.789 at 60 F. Commercial alcohol always contains from 10 to 40 per cent water. Latent heat at low pressures can be assumed as 400 Btu/lb and specific heat as 0.65 Btu/lb-°F.

In several European countries and in Japan, which have to import gasoline, the law requires that a certain amount of alcohol, from 3.2 to 20 per cent, depending upon the country, be added to gasoline when sold as motor fuel. The main drawback to this requirement is that gasoline mixes only with water-free alcohol, which is more difficult to produce. Another disadvantage is that the cost of alcohol is higher than that of gasoline.

Methyl, or wood, alcohol, CH_4O, is now produced chiefly synthetically. Its specific gravity is 0.796 at 60 F, latent heat about 510 Btu/lb and specific heat 0.60 Btu/lb.

Denatured alcohol is obtained by adding to 100 parts of ethyl alcohol by volume either 10 parts of methyl alcohol and ½ part of benzene or 2 parts of methyl alcohol and ½ part of pyridine—a heavy hydrocarbon with a distinctive color and odor.

Vapor-pressure curves of both alcohols are given in Fig. 3-10.

Example 3-6.—Determine the ratio r_{av} of air to saturated ethyl alcohol vapor for a mixture temperature of 70 F.

From Fig. 3-10 the vapor pressure at 70 F is 0.98 psia; the partial pressure of air is $14.70 - 0.98 = 13.72$ psia.

Specific weight of air is from the characteristic equation

$$w_a = 144p/RT = 144 \times 13.72/[53.34 \times (70 + 460)] = 0.0698 \text{ lb/cu ft}$$

Similarly the specific weight of saturated alcohol vapor is found from the equation

$$w_v = 144 \times 0.98/[(1545/46) \times 530] = 0.00793 \text{ lb/cu ft}$$

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[1] *Automotive Industries*, vol. 77, No. 22, pp. 784–787, Nov. 27, 1937; *Diesel Power*, September 1940, pp. 767–779.

[2] VOGT, C. J., Tests Show Merits of Butane as an Internal-combustion Engine Fuel, *Automotive Industries*, vol. 71, pp. 348–351, Sept. 22, 1934.

TATTERSFIELD, E. E., Development of Butane as Motor-truck Fuel, *SAE Journal*, vol. 47, No. 2, p. 311, August, 1940.

Therefore

$$r_{av} = 0.0698/0.00793 = 8.8 \text{ lb air per lb vapor}$$

Example 3-7.—Determine the temperature to which air of 70 F and 14.0 psi pressure must be heated for 100 per cent evaporation for an air-fuel ratio $r_{af} = 6.50$ when methyl alcohol is the fuel.

From equation (3-9) the partial pressure of alcohol vapor is

$$p_f = 14.0/(1 + 6.50 \times 32/29) = 1.714 \text{ psia}$$

or

$$1.714/0.491 = 3.49 \text{ in. Hg}$$

For this pressure Fig. 3-10 gives a temperature of 65 F.

From equation (3-10) heat which must be supplied per pound of alcohol

$$Q = (6.50 \times 0.24 + 0.60)(65 - 70) + 510 \times 1.0 = 499.2 \text{ Btu/lb}$$

The temperature to which 6.5 lb of air must be heated to supply this heat is

$$t = t_1 + Q/(r_{af}c_p) = 70 + 499.2/(6.5 \times 0.24) = 390 \text{ F}$$

The last example shows also that the temperature to which air must be preheated is rather high on account of the high value of latent heat of alcohol.

Detonation.—Methyl alcohol behaves about the same as average gasoline and has a maximum compression ratio of 5.2. Ethyl alcohol does not detonate even with a compression ratio of 7.5. Added to gasoline, ethyl alcohol has an antiknock effect, 11 per cent of it stopping detonation under average conditions.

3-7. Problems—1. Find the temperature required to produce a 16:1 air-vapor mixture for 60 per cent evaporated fuel for gasoline having volatility characteristics shown in Fig. 3-6.

2. Determine graphically and analytically for the gasoline in problem 1 what part must be evaporated to obtain a 18:1 air-vapor mixture from a 4:1 air-fuel mixture.

3. Find the lowest temperature at which an engine can be started using fuel *C*, Fig. 3-5, if its carburetor can produce an air-fuel mixture not richer than 1.6:1.

4. If the carburetor in problem 3 can be adjusted to give a mixture 1.2:1, how will this influence the starting temperature?

5. Find the dew point for a 15:1 mixture of United States fighting-grade aviation gasoline at a pressure of 13.5 psia.

6. Determine the heat necessary to obtain 65 per cent vaporization in the intake manifold for an outside temperature of 68 F and 1 psi below atmospheric pressure, using gasoline *A*, Fig. 3-5.

7. Determine the amount of additional heat necessary for complete vaporization in problem 6.

8. Determine the temperature to which air of 68 F must be heated to fulfill the conditions of problem 7.

9. Find the gravity in degrees API for an oil having a specific gravity of 0.84.

10. Find the specific gravity of a fuel oil which has a gravity of 28°API.

11. Find the specific gravity of kerosene having a gravity of 45°API.

12. Determine the partial pressure of octane for an air-vapor ratio 18:1 and a carburetor pressure of 14.0 psia.

13. Determine the temperature to which air of 80 F and 13.95 psia pressure must be heated for 80 per cent evaporation from a $r_{af} = 15$, using hexane as fuel.

14. Determine the temperatures to which air of 72 F and 14.1 psi pressure must be heated to obtain 80 and 100 per cent vaporization for an air-fuel ratio 16:1, using kerosene as fuel.

15. Determine the ratio r_{av} of air to saturated methyl alcohol vapor for a mixture temperature of 65 F and pressure of 13.2 psia in the carburetor.

16. Determine the temperature to which air of 65 F and 12.9 psia pressure must be heated for complete evaporation, using a fuel mixture of 80 per cent ethyl alcohol and 20 per cent methyl alcohol and air-fuel ratios, 9.03 for C_2H_6O and 6.50 for CH_4O.

17. Compute and present by graphs the relation between octane and cetane rating of fuels.

CHAPTER 4

COMBUSTION

4-1. Definition.—Combustion is a chemical reaction in which certain elements of the fuel combine with oxygen causing an increase in temperature of the gases. The main combustible elements are carbon and hydrogen; another combustible element often present in fuels, although rather undesirable, is sulphur.

The oxygen necessary for combustion is obtained from air, which is oxygen diluted chiefly by nitrogen.

Computations related to combustion are carried either in units of weight or in units of volume, and Table 4-1 gives proportions of oxygen and nitrogen for both cases.

TABLE 4-1.—COMPOSITION OF ATMOSPHERIC AIR

Constituents	Symbol	Mol. wt. M	Analysis, per cent		Relative to O_2		Mol. wt. per mol air
			By vol.[a]	By wt.	Vol.	Wt.	
Oxygen..........	O_2	32.0	20.99	23.2	1	1	6.717
Nitrogen........	N_2	28.02	78.03	76.8	3.76	3.31	21.848
Argon...........	A	40.0	0.94				0.376
Carbon dioxide...	CO_2	44.0	0.03				0.013
Other gases......	....	.....	0.01				
Total air......	....	28.95	100.0	100.0	4.76	4.31	28.95

[a] International Critical Tables, vol. 1, p. 393, McGraw-Hill Book Company, Inc., New York, 1926.

4-2. Combustion Formulas.—The amounts of the necessary oxygen or air and of the resultant products of combustion may be calculated by the use of chemical equations called *combustion formulas.*

Carbon.—In combining with oxygen, carbon forms carbon dioxide, but under certain conditions carbon monoxide may also be formed.

Complete combustion of carbon to carbon dioxide is expressed by the chemical equation

$$C + O_2 = CO_2 \qquad (4\text{-}1)$$

which shows that 1 molecule of carbon combines with 1 molecule of oxygen and forms 1 molecule of carbon dioxide.

In the mol system, equation (4-1) is read as

$$1 \text{ mol } C + 1 \text{ mol } O_2 = 1 \text{ mol } CO_2 \qquad (4\text{-}2)$$

Substituting for each molecule its molecular weight gives

$$12 + 32 = 44 \qquad (4\text{-}3)$$

This equation shows that 12 units of weight of carbon combine with 32 units of oxygen and form 44 units of carbon dioxide. For 1 lb carbon, dividing equation (4-3) by 12 gives

$$1 \text{ lb } C + 2.67 \text{ lb } O_2 = 3.67 \text{ lb } CO_2 \qquad (4\text{-}4)$$

For air, each pound O_2 being accompanied by 3.31 lb N_2, equation (4-4) becomes

$$\boxed{1 \text{ lb } C + 2.67 \text{ lb } O_2 + 8.84 \text{ lb } N_2 = 3.67 \text{ lb } CO_2 + 8.84 \text{ lb } N_2} \qquad (4\text{-}5)$$

indicating that 1 lb carbon requires 2.67 + 8.84 = 11.51 lb air.

For air, molal equation (4-2) must be changed by adding 3.76 mol of N_2 for each mol of O_2:

$$1 \text{ mol } C + 1 \text{ mol } O_2 + 3.76 \text{ mol } N_2 = 1 \text{ mol } CO_2 + 3.76 \text{ mol } N_2 \qquad (4\text{-}6)$$

Substituting for carbon the value of its mol in pounds, $M = 12$, and for the gases the value of mol in cubic feet at standard conditions and dividing by 12 give

$$\begin{aligned} &1 \text{ lb } C + 31.6 \text{ cu ft } O_2 + 118.8 \text{ cu ft } N_2 \\ &\qquad = 31.6 \text{ cu ft } CO_2 + 118.8 \text{ cu ft } N_2 \qquad (4\text{-}7) \end{aligned}$$

This indicates that 1 lb carbon requires 31.6 + 118.8 = 150.4 cu ft air at standard conditions.

If combustion is conducted with an excess of air, e being the excess in cubic feet per 1 cu ft of necessary air, equation (4-7) becomes

$$\begin{aligned} &1 \text{ lb } C + (1 + e)150.4 \text{ cu ft air} \\ &= 31.6 \text{ cu ft } CO_2 + 31.6e \text{ cu ft } O_2 + (1 + e)118.8 \text{ cu ft } N_2 \qquad (4\text{-}8) \end{aligned}$$

Carbon Monoxide.—If the result of combustion of C is carbon monoxide, CO, the molecular formula is

$$1 \text{ mol C} + \tfrac{1}{2} \text{ mol } O_2 + 1.88 \text{ mol } N_2 = 1 \text{ mol CO} + 1.88 \text{ mol } N_2 \quad (4\text{-}9)$$

Proceeding as above, this formula can be presented as

$$1 \text{ lb C} + 15.8 \text{ cu ft } O_2 + 59.4 \text{ cu ft } N_2 \\ = 31.6 \text{ cu ft CO} + 59.4 \text{ cu ft } N_2 \quad (4\text{-}10)$$

indicating that 1 lb carbon when forming carbon monoxide uses $15.8 + 59.4 = 75.2$ cu ft air.

Combustion of CO to CO_2 can be expressed by the equation

$$1 \text{ mol CO} + \tfrac{1}{2} \text{ mol } O_2 + 1.88 \text{ mol } N_2 \\ = 1 \text{ mol } CO_2 + 1.88 \text{ mol } N_2 \quad (4\text{-}11)$$

All parts being gases with the same value for mol,

$$1 \text{ cu ft CO} + 0.5 \text{ cu ft } O_2 + 1.88 \text{ cu ft } N_2 \\ = 1 \text{ cu ft } CO_2 + 1.88 \text{ cu ft } N_2 \quad (4\text{-}12)$$

indicating that 1 cu ft CO requires $0.50 + 1.88 = 2.38$ cu ft air.

Combustion with excess e of air:

$$1 \text{ cu ft CO} + (1 + e)2.38 \text{ cu ft air} \\ = 1 \text{ cu ft } CO_2 + 0.5e \text{ cu ft } O_2 + (1 + e)1.88 \text{ cu ft } N_2 \quad (4\text{-}13)$$

Hydrogen.—When H_2 is burned, it produces H_2O. The molecular equation of complete combustion can be written as

$$1 \text{ mol } H_2 + \tfrac{1}{2} \text{ mol } O_2 + 1.88 \text{ mol } N_2 \\ = 1 \text{ mol } H_2O + 1.88 \text{ mol } N_2 \quad (4\text{-}14)$$

In changing equation (4-14) to cubic feet it should be remembered that H_2O at standard conditions becomes a liquid; the part left as vapor is less than 2 per cent of each mol and may be disregarded.

Substituting for the mols for gases 379 cu ft and for H_2O its weight 18 lb and dividing by 379 gives

$$1 \text{ cu ft } H_2 + 0.5 \text{ cu ft } O_2 + 1.88 \text{ cu ft } N_2 \\ = 0.0475 \text{ lb } H_2O + 1.88 \text{ cu ft } N_2 \quad (4\text{-}15)$$

Sulphur, S, burns usually to sulphur dioxide, SO_2, a gas at standard conditions. The molecular equation is

$$1 \text{ mol S} + 1 \text{ mol O}_2 + 3.76 \text{ mol N}_2 = 1 \text{ mol SO}_2 + 3.76 \text{ mol N}_2 \qquad (4\text{-}16)$$

Substituting the values for mols and referring to 1 lb sulphur gives

$$1 \text{ lb S} + 56.4 \text{ cu ft air} = 11.8 \text{ cu ft SO}_2 + 44.5 \text{ cu ft N}_2 \qquad (4\text{-}17)$$

Hydrogen sulphide, H_2S, is a gas sometimes present in natural gas. It burns to water and sulphur dioxide, and the chemical equations may be written and computations made as though hydrogen and sulphur were separate:

$$1 \text{ mol H}_2\text{S} + 1.5 \text{ mol O}_2 + 5.64 \text{ mol N}_2 = 1 \text{ mol H}_2\text{O} + 1 \text{ mol SO}_2 + 5.64 \text{ mol N}_2 \qquad (4\text{-}18)$$

Substituting for 1 mol of the gases 379 cu ft and for 1 mol water 18 lb and dividing by 379 gives

$$1 \text{ cu ft H}_2\text{S} + 7.14 \text{ cu ft air} = 0.0475 \text{ lb H}_2\text{O} + 1 \text{ cu ft SO}_2 + 5.64 \text{ cu ft N}_2 \qquad (4\text{-}19)$$

Compound Gases.—Air required for the combustion of 1 cu ft of a compound gas may be computed by writing the combustion equation for each constituent and the air required per cubic foot of the constituent. These values are then multiplied by the percentages by volume of the different constituents in the mixture and the products added, the total giving cubic feet of air required per cubic foot of the compound gas.

4-3. Formulas for Hydrocarbons.—All hydrocarbons whose general chemical formula is C_nH_m burn, in the presence of a sufficient amount of oxygen, to carbon dioxide and water, and their combustion equations may be written and all computations made as though carbon and hydrogen were separate.

In the following, formulas for two representative compounds will be deduced to illustrate the above statement, and then the general equations will be given.

Methane (marsh gas), CH_4.—The molecular equation of combustion in air is

$$1 \text{ mol CH}_4 + 2 \text{ mol O}_2 + 7.52 \text{ mol N}_2 = 1 \text{ mol CO}_2 + 2 \text{ mol H}_2\text{O} + 7.52 \text{ mol N}_2 \qquad (4\text{-}20)$$

Changing this equation to cubic feet but expressing H_2O in pounds gives

$$\begin{aligned}&1 \text{ cu ft } CH_4 + 9.52 \text{ cu ft air}\\ &\qquad = 1 \text{ cu ft } CO_2 + 0.095 \text{ lb } H_2O + 7.52 \text{ cu ft } N_2 \qquad (4\text{-}21)\end{aligned}$$

Benzene, C_6H_6, is a liquid; its molecular equation of combustion in air is

$$\begin{aligned}&1 \text{ mol } C_6H_6 + 7.5 \text{ mol } O_2 + 28.2 \text{ mol } N_2\\ &\qquad = 6 \text{ mol } CO_2 + 3 \text{ mol } H_2O + 28.2 \text{ mol } N_2 \qquad (4\text{-}22)\end{aligned}$$

Substituting the values of mol in pounds for the liquids and in cubic feet for gases and referring to 1 lb C_6H_6 gives

$$\begin{aligned}&1 \text{ lb } C_6H_6 + 173.3 \text{ cu ft air}\\ &\qquad = 29.2 \text{ cu ft } CO_2 + 0.69 \text{ lb } H_2O + 137.0 \text{ cu ft } N_2 \qquad (4\text{-}23)\end{aligned}$$

General equation for combustion of a hydrocarbon, C_nH_m, with an excess of air e, may be written as follows:

$$\begin{aligned}&1 \text{ mol } C_nH_m + (1 + e)(n + 0.25m) \text{ mol } O_2\\ &\qquad + 3.76(1 + e)(n + 0.25m) \text{ mol } N_2\\ &\quad = n \text{ mol } CO_2 + 0.5m \text{ mol } H_2O + e(n + 0.25m) \text{ mol } O_2\\ &\qquad\qquad + 3.76(1 + e)(n + 0.25m) \text{ mol } N_2 \qquad (4\text{-}24)\end{aligned}$$

Depending upon the state of the hydrocarbon, whether it is a liquid or a gas, equation (4-24) can be presented in two different forms.

For a hydrocarbonous gas

$$\begin{aligned}&1 \text{ cu ft } C_nH_m + 4.76(1 + e)(n + 0.25m) \text{ cu ft air}\\ &\quad = n \text{ cu ft } CO_2 + 0.0238m \text{ lb } H_2O + e(n + 0.25m) \text{ cu ft } O_2\\ &\qquad\qquad + 3.76(1 + e)(n + 0.25m) \text{ cu ft } N_2 \qquad (4\text{-}25)\end{aligned}$$

For a liquid hydrocarbon a more suitable form is

$$\begin{aligned}&1 \text{ lb } C_nH_m + \frac{1804(1 + e)(n + 0.25m)}{12n + m} \text{ cu ft air}\\ &= \frac{379n}{12n+m} \text{ cu ft } CO_2 + \frac{9m}{12n+m} \text{ lb } H_2O + \frac{379e(n+0.25m)}{12n + m} \text{ cu ft } O_2\\ &\qquad + \frac{1426(1 + e)(n + 0.25m)}{12n + m} \text{ cu ft } N_2 \qquad (4\text{-}26)\end{aligned}$$

Complete combustion with theoretical air can be considered as a special case with $e = 0$ in the last three equations.

Alcohols.—Both alcohols available as engine fuel, *i.e.*, ethyl alcohol, C_2H_6O, and methyl alcohol, CH_4O, burn to carbon dioxide and water. Their combustion equations are similar to those of hydrocarbons. The atom of oxygen in their molecule can be considered as though it is already combined with 1 molecule of H_2, leaving for combustion a true hydrocarbon. Thus for ethyl alcohol, $C_2H_6O = H_2O + C_2H_4$; and for methyl alcohol, $CH_4O = H_2O + CH_2$.

Ethyl Alcohol.—For combustion of C_2H_6O in air

$$\begin{aligned} &1 \text{ mol } C_2H_6O + 3 \text{ mol } O_2 + 11.28 \text{ mol } N_2 \\ &\qquad = 2 \text{ mol } CO_2 + 3 \text{ mol } H_2O + 11.28 \text{ mol } N_2 \end{aligned} \tag{4-27}$$

which gives for 1 lb alcohol

$$\begin{aligned} &1 \text{ lb } C_2H_6O + 117.7 \text{ cu ft air} \\ &\qquad = 14.6 \text{ cu ft } CO_2 + 1.17 \text{ lb } H_2O + 92.7 \text{ cu ft } N_2 \end{aligned} \tag{4-28}$$

Methyl Alcohol.—For combustion of CH_4O in air

$$\begin{aligned} &1 \text{ mol } CH_4O + 1.5 \text{ mol } O_2 + 5.64 \text{ mol } N_2 \\ &\qquad = 1 \text{ mol } CO_2 + 2 \text{ mol } H_2O + 5.64 \text{ mol } N_2 \end{aligned} \tag{4-29}$$

which gives for 1 lb alcohol

$$\begin{aligned} &1 \text{ lb } CH_4O + 84.6 \text{ cu ft air} \\ &\qquad = 11.8 \text{ cu ft } CO_2 + 1.13 \text{ lb } H_2O + 66.8 \text{ cu ft } N_2 \end{aligned} \tag{4-30}$$

4-4. Air-fuel Ratio.—The number of pounds of air present in the engine cylinder per pound of fuel is called *air-fuel ratio* and is designated as r_{af}. Theoretically the air-fuel ratio necessary for complete combustion depends only upon the composition of the fuel; practically it depends also upon how thoroughly the air and fuel are mixed so that their particles can combine properly.

The air-fuel ratio can be found from the above given combustion formulas. For instance, for a hydrocarbon C_nH_m, equation (4-26) gives directly the volume of air per pound of hydrocarbon; multiplying the volume of air by its specific weight at standard conditions, $w = 0.0763$, the weight of air representing the air-fuel ratio can be written as

$$r_{af} = \frac{137.6(n + 0.25m)(1 + e)}{12n + m} \tag{4-31}$$

When testing internal-combustion engines the air-fuel ratio can be determined on the basis of the exhaust-gas analysis. Designating by CO_2, O_2, and N_2 the percentages of these gases as found by the analysis and noticing that H_2O is condensed and does not appear in the volumetric analysis gives for the percentage of CO_2 from the right side of equation (4-24)

$$CO_2 = \frac{100n}{n + e(n + 0.25m) + 3.76(1 + e)(n + 0.25m)} \qquad (4\text{-}32)$$

Dividing every term of this expression by n gives

$$CO_2 = \frac{100}{1 + e(1 + 0.25m/n) + 3.76(1 + e)(1 + 0.25m/n)} \qquad (4\text{-}33)$$

The ratio m/n for the hydrocarbon C_nH_m used by the engine may not be known, but the ultimate analysis of the fuel gives the percentage by weight of carbon, C, and hydrogen, H, and it is easy to see that

$$H/C = m/12n \qquad (4\text{-}34)$$

Substituting 12 H/C for m/n in equation (4-33) and solving for e give

$$e = \frac{100 - (4.76 + 11.34H/C)CO_2}{(4.76 + 14.34H/C)CO_2} \qquad (4\text{-}35)$$

For practical use, expression (4-31) may be altered using relation (4-34), which gives

$$r_{af} = \frac{11.53(1 + 3H/C)(1 + e)}{1 + H/C} \qquad (4\text{-}36)$$

Equations similar to equation (4-35) can be written based on other components of the exhaust-gas analysis, O_2 and N_2, and e determined as a function of O_2 or N_2 instead of CO_2. However, the values of r_{af} thus computed are less reliable than those from equation (4-35) on account of greater probable errors of determination of O_2 and especially of N_2 by the conventional exhaust-gas analysis.

In actual engine performance the combustion usually is not complete even in the presence of excess air, because of dissociation at high temperatures and also unsatisfactory mixing of the air and the fuel. In such a case the exhaust gases may contain certain percentages of CO, H_2, NO, OH, H, O, and CH_4. By reasoning similar to that used for deducing equations (4-24), and (4-15), proper equations may be found for computing e and r_{af} under these conditions.[1]

4-5. Products of Combustion. *Gas Analysis.*—An analysis obtained with a regulation Orsat apparatus gives much useful information. However, it must be borne in mind that such an analysis gives the composition of exhaust gases on a water-free basis and therefore is not the true analysis of the products.

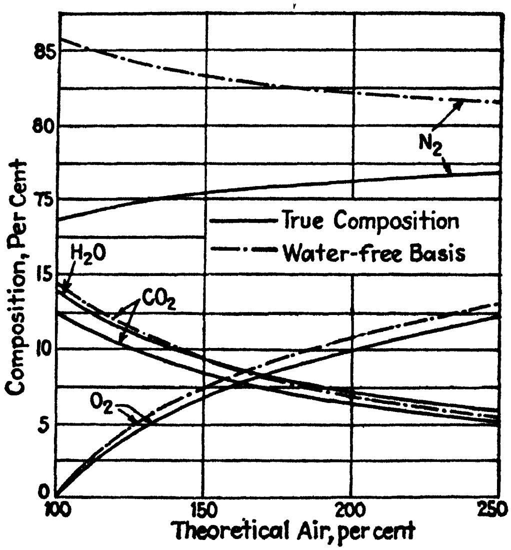

FIG. 4-1.—Products of combustion of gasoline.

The analysis on the water-free basis differs from the true analysis, particularly in regard to N_2. With the increase of excess air the percentage of N_2 increases, whereas the water-free basis shows a gradual decrease. These conditions are illustrated for combustion of gasoline in Fig. 4-1. The divergence between the true and the water-free analyses is very marked, particularly with the smaller amounts of air.

Example 4-1.—Determine the exhaust-gas analysis for complete combustion of kerosene, $\gamma = 0.79$, with theoretical air.

If light kerosene, $C_{10}H_{22}$, is used for this, the right side of equation (4-24) with $e = 0$ gives

$$10 \text{ mol } CO_2 + 0.5 \times 22 \text{ mol } H_2O + 3.76(10 + 0.25 \times 22) \text{ mol } N_2$$
$$= 10 \text{ mol } CO_2 + 11 \text{ mol } H_2O + 58.3 \text{ mol } N_2$$

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[1] MALEEV, V. L., Air-fuel Ratio from Exhaust-gas Analysis of Oil Engines, *Bull. Eng. Exp. Sta., Oklahoma A. & M. Coll.*, vol. 2, No. 5, 1931.

This gives for the theoretical (imaginary) analysis

$CO_2 = 10/(10 + 11 + 58.3) = 0.1261$ or...	12.61 per cent
$H_2O = 11/79.3 = 0.1387$ or................	13.87
$N_2 = 58.3/79.3 = 0.7352$ or.............	73.52
Total................................	100.00 per cent

Determined by an Orsat apparatus the water-free analysis would be

$CO_2 = 10/(10 + 58.3) = 0.146$ or...........	14.6 per cent
$N_2 = 58.3/68.3 = 0.854$ or.................	85.4 per cent

Volume of the products of combustion for a fuel which is a mixture of combustible gases or liquids can be computed from the above given combustion formulas by first finding the volumes of the constituents of the products, CO_2, N_2, and O_2, if excess air was used, for each constituent of the fuel, then by multiplying the volumes of products formed per unit of each combustible by its fraction in the fuel and by the ratio of temperatures $(t + 460)/520$, and finally adding these corrected volumes together. If t is lower than the saturation temperature of water vapor corresponding to its partial pressure p_w in the products, the volume of H_2O can be neglected. If t is higher than the saturation temperature, the volume of H_2O must be added as superheated steam of temperature t and pressure p_w.

Change of Volume.—As a result of chemical reactions, the volume of the products of combustion usually differs from the volume of the combustible mixture, both taken at the same pressure and temperature. Sometimes there is an increase in volume and sometimes a shrinkage of volume. The volume of 1 mol being the same for all gases, the change of volume is best determined by the change of mols in the combustion equation. For CO, equation (4-11) shows that 3.38 mols CO, O_2, and N_2 shrink to 2.88 mols CO_2 and N_2, or a decrease of 14.8 per cent.

Methane burns without a change of volume, equation (4-20), whereas combustion of benzene, equation (4-22), results in a change of volume $(6 + 3 + 28.2) \div (1 + 7.5 + 28.2) = 1.013$, or 1.3 per cent increase.

The presence of excess air decreases the volume change.

Gas Constant.—A change of volume causes a corresponding change in the gas constant, as can be seen comparing the characteristic equations at the same pressures and temperatures before and after combustion

$$pV_1 = WR_1T \qquad \text{and} \qquad pV_2 = WR_2T$$

Dividing the second by the first gives

$$R_2 = R_1V_2/V_1 \tag{4-37}$$

or designating the numbers of mols as found from the combustion equations by n_1 and n_2 gives

$$R_2 = R_1n_2/n_1 \tag{4-38}$$

Generally, compound gaseous fuels give a decrease of volume and $R_2 < R_1$, whereas liquid fuels give an increase of volume and $R_2 > R_1$. The change of the gas constant usually is not over ±6 per cent.

Example 4-2.—Determine the change of the gas constant when burning gasoline with 10 per cent excess air.

For gasoline, C_8H_{18}, $n = 8$, $m = 18$, and from equation (4-24)

$$n_1 = 1 + (1 + 0.1)(8 + 0.25 \times 18) + 3.76(1 + 0.1)(8 + 0.25 \times 18) = 66.45$$
$$n_2 = 8 + 0.5 \times 18 + 0.1 \times 12.5 + 3.76 \times 1.1 \times 12.5 = 69.95$$
$$R_2 = R_1 \times 69.95/66.45 = 1.053R_1$$

or an increase of 5.3 per cent.

Partial pressures of the constituent gases of the products of combustion can be determined from equation (2-13) or (2-14), depending upon whether the composition of the exhaust gases is known in units of volume or of weight.

Example 4-3.—Find the partial pressures of the products formed by gasoline burned with 10 per cent excess air.

Since 1 mol of any gas at a given pressure and temperature occupies the same volume, the ratios of volumes are equal to the ratios of mols. Therefore, taking the figures from Example 4-2:

For CO_2, $r_{v1} = 8/69.95 = 0.1144$ and from equation (2-13)

$$p_1 = 14.7 \times 0.1144 = 1.682 \text{ psi}$$

for H_2O, pressure $p_2 = 14.7 \times 9/69.95 = 1.892$ psi;

for O_2, pressure $p_3 = 14.7 \times 1.25/69.95 = 0.262$ psi;

and, finally, for N_2, pressure $p_4 = 14.7 \times 51.70/69.95 = 10.864$ psi. A check by equation (2-12) gives $1.682 + 1.892 + 0.262 + 10.864 = 14.70$.

The partial pressure of H_2O determines the temperature of saturation at this pressure or the temperature at which the water vapor will begin to condense. Interpolating from steam tables for $p_2 = 1.892$ gives $t = 123.6$ F

Example 4-4.—Determine the imaginary and actual volumes at 75 F of the products of combustion per pound of fuel in Example 4-3.

Similar to equation (4-26), the total or imaginary volume at t F is

$$V = [n_2 \times 379/(12n + m)](t + 460)/520 \qquad (4\text{-}39)$$

or with the data from the previous example

$$V = [69.95 \times 379/(12 \times 8 + 18)](75 + 460)/520 = 239.3 \text{ cu ft}$$

The partial pressure of water vapor at 75 F from the steam tables is 0 430 psi. This leaves as the partial pressure of the other constituents

$$14.70 - 0.43 = 14.27 \text{ psi}$$

The partial pressure of the remaining constituents before condensation of water vapor is, from Example 4-3,

$$1.682 + 0.262 + 10.864 = 12.808 \text{ psi}$$

The increase of partial pressure is accompanied by a decrease of volume in inverse proportion. Therefore the actual volume is

$$239.3 \times 12.81/14.27 = 214.8 \text{ cu ft}$$

4-6. Heat Values.—The chemical reaction between a combustible and oxygen produces heat. The heat thus generated when one unit of combustible is completely burned is called the *heating value* or *heat value* of the combustible.

Higher Heat Value.—Heat values are determined experimentally by calorimeters. In these the products of combustion are cooled to practically atmospheric temperature at which most of the water vapor from combustion of hydrogen is condensed, giving up its latent heat. The heat value thus determined is called *higher heat value* and designated Q_h.

Lower Heat Value.—Calculations of the temperature attained by combustion are facilitated by the use of a so-called *lower heat value* Q_l, which eliminates the necessity of taking into account the latent heat of the water vapor and the difference between the specific heats of liquid water and of water vapor.

The quantity which must be subtracted from the higher heat value Q_h to obtain the lower heat value Q_l varies with the composition of the fuel; an approximate value is $1050w$, where 1050 is the latent heat of evaporation of 1 lb water at 77 F, the accepted calorimeter temperature; w is the number of pounds of H_2O formed per pound or cubic foot of fuel burned; w is equal to 9(H) if (H) designates the part by weight of hydrogen in the fuel; thus

$$Q_l = Q_h - 9450(\mathrm{H}) \tag{4-40}$$

In calculating the thermal efficiency of internal-combustion engines it would be logical to use the lower heat values because of the high temperature of the exhaust gases leaving the engine. However, the ASME Power Test Code specifies the use of the higher heat values. Actually it does not make any difference so long as it is clearly stated which one of the heat values is used.

Heat Values for Liquids and Gases.—Usually the heat value is determined in Btu per pound. For gaseous fuels it is more convenient to have the heat value determined in Btu per cubic foot at standard pressure and temperature. The heat value per cubic foot may be found by multiplying the heat value expressed in Btu per pound by the density of the gas. Or, if the heat value per mol is known, the heat value per cubic foot may be found by dividing the Btu per mol by the volume of the gas per mol, or 379 for standard conditions.

Heat-value Formulas.—If the chemical composition of a fuel is known, its heat value may be calculated fairly closely. The formula is based on heat values of the combustible elements, carbon, hydrogen, and sulphur.

The following modification of Dulong's formula, which takes into consideration the heat necessary to break up the chemical union of the carbon and hydrogen, by an average value of −1150 Btu/lb, and the small percentage of sulphur usually present in oils, may be used for hydrocarbons, both gaseous and liquid, and for alcohols, and gives values in Btu per pound.

$$Q_h = 14{,}520\ \mathrm{C} + 61{,}045(\mathrm{H} - \mathrm{O}/8) + 3982\ \mathrm{S} - 1150 \tag{4-41}$$

For liquid fuels of unknown ultimate analysis special formulas, based on their gravity in degrees API, may be used.

For gasoline, also in Btu per pound,

$$Q_h = 17{,}900 + 40 \times {}^\circ\mathrm{API} \tag{4-42}$$
$$Q_l = 16{,}610 + 40 \times {}^\circ\mathrm{API} \tag{4-43}$$

For kerosene:

$$Q_h = 18{,}030 + 40 \times {}^\circ\mathrm{API} \tag{4-44}$$
$$Q_l = 16{,}780 + 40 \times {}^\circ\mathrm{API} \tag{4-45}$$

TABLE 4-2.—FUEL DATA
(Volumes at 14.7 psia and 60 F)

Fuel	Chemical symbol	Approx. mol. wt.	Boiling point at 14.7 psia, °F	Volume, cu ft			Air fuel ratio with $e = 0$	Heat values, Btu			
				Of 1 lb fuel	Of theoretical air			Higher		Lower	
					Per cu ft fuel	Per lb fuel		Per lb	Per cu ft	Per lb	Per cu ft
Hydrogen	H_2	2	−423	189.0	2.39	450.0	34.63	61,045	324.5	51,608	274
Carbon[a] to CO	C	12	.....	(31.6)	2.39	75.2	5.77	4,340	137.5	4,340	138
Carbon[a] to CO_2	C	12	.....	(31.6)	4.75	150.3	11.56	14,520	458.5	14,520	458
Carbon monoxide	CO	28	−314	13.60	2.39	32.2	2.48	4,345	320.7	4,345	321
Sulphur[a]	S	32	+832	(11.87)	4.76	56.4	4.31	3,982	335.5	3,982	336
Hydrogen sulphide	H_2S	34	−75	11.18	7.14	79.6	6.09	7,000	628	6,500	583
Methane	CH_4	16	−258	23.75	9.52	225.5	17.32	23,910	1,011	21,530	910
Ethane	C_2H_6	30	−127	12.62	16.65	210.0	16.13	22,330	1,757	20,425	1,633
Propane	C_3H_8	44	−44	8.61	23.8	205.0	15.65	21,670	2,515	19,940	2,332
Butane	C_4H_{10}	58	+14	6.52	30.9	201.5	15.45	21,320	3,270	19,670	3,036
Pentane[b]	C_5H_{12}	72	+97	5.26	38.0	199.8	15.30	21,100	4,000	19,514	3,718
Hexane[b]	C_6H_{14}	86	+156	4.40	45.2	198.8	15.30	20,950	4,730	19,400	4,408
n-Heptane[b]	C_7H_{16}	100	+200	3.78	52.4	198.2	15.15	20,800	5,475	19,290	5,075
n-Octane (gasoline)[b]	C_8H_{18}	114	+256	3.32	59.5	197.5	15.19	20,750	6,210	19,250	5,810
Decane (paraffin)[b]	$C_{10}H_{22}$	142	+345	2.67	73.8	197.0	15.03	20,500	7,680	19,150	7,170
n-Dodecane (kerosene)[b]	$C_{12}H_{26}$	170	+421	2.23	88.0	196.2	15.07	20,440	9,170	19,110	8,570
Ethylene	C_2H_4	28	−155	13.52	14.28	192.9	14.85	21,660	1,600	20,300	1,515
Propylene	C_3H_6	42	−53	9.02	21.4	192.9	14.85	21,330	2,335	19,970	2,197
Acetylene	C_2H_2	26	−118	14.57	11.90	173.5	13.32	20,880	1,477	20,770	1,436
Benzene[b]	C_6H_6	78	+175	4.86	35.7	173.5	13.32	18,160	3,740	17,190	3,600
Toluene[b]	C_7H_8	92	+231	4.12	42.8	176.2	13.50	18,300	4,440	17,460	4,260
Hexadecadiene (fuel oil)[b]	$C_{16}H_{30}$	222	+500	(1.7)	(112.0)	190.6	14.58	19,500	(11,500)	18,225	(10,760)
Methyl alcohol[b]	CH_4O	32	+147	11.83	7.14	84.4	6.49	10,270	866.2	9,078	765
Ethyl alcohol[b]	C_2H_6O	46	+173	8.23	14.28	117.5	9.03	13,170	1,597	11,930	1,446

[a] Assumed to be gasified.
[b] Assumed to be vapor.

For fuel oil (ASME formula):

$$Q_h = 17{,}680 + 60 \times {}^\circ\mathrm{API} \tag{4-46}$$

which gives in accordance with equations (4-42) to (4-45)

$$Q_l = 16{,}380 + 60 \times {}^\circ\mathrm{API} \tag{4-47}$$

Fuel Data.—Computations pertaining to combustion and heat value are facilitated by the use of data given in Table 4-2. This is particularly true in the case of compound gases.

Example 4-5.—A fuel is used which is called methylated spirits and which is a mixture of 90 per cent by volume ethyl alcohol and 10 per cent methyl alcohol. Determine the volume of air per pound of this fuel with 20 per cent excess air at 14.7 psi and 150 F.

The volumetric relation must be changed to weight relation. The specific gravity of C_2H_6O being 0.789 and that of CH_4O being 0.796, their proportion by weight will be:

For C_2H_6O:

$$0.90 \times 0.789/(0.789 \times 0.90 + 0.796 \times 0.10) = 0.8993 \text{ or } 89.93 \text{ per cent}$$

For CH_4O:

$$0.10 \times 0.796/(0.789 \times 0.90 + 0.796 \times 0.10) = 0.1007 \text{ or } 10.07 \text{ per cent}$$

The volume of air, using Table 4-2, is

$$V = (0.8993 \times 117.5 + 0.1007 \times 84.4) \times (1 + 0.20) = 136.8 \text{ cu ft}$$

Taking the volumes of the fuel from Table 4-2, adding them to the volume of the air, and referring the sum to 150 F give

$$V = (0.8993 \times 8.23 + 0.1007 \times 11.83 + 136.8)(150 + 460)/520 = 170.5 \text{ cu ft}$$

Example 4-6.—Find the higher heat value per cubic foot of the above mixture.

From Table 4-2,

$$Q_h = 0.8993 \times 13{,}170 + 0.1007 \times 10{,}270 = 12{,}878 \text{ Btu/lb}$$

and per cubic foot of air-fuel mixture

$$Q_h = 12{,}878/170.5 = 75.5 \text{ Btu/cu ft}$$

4-7. Additional Data. *Chemical Energy.*—The chemical energy C and the heat value of a fuel Q, as determined by a calorimeter test, are not the same thing. Although the numerical difference may be small, there is a distinct difference in the concepts. The heat value is determined either in a constant-volume or bomb calorimeter and designated Q_v, or for gaseous and volatile liquids, it is determined in a constant-pressure continuous-burning calorimeter and designated Q_p. In both cases the products of combustion are cooled to the initial temperature.

For the first case the energy equation for the conditions before and after the combustion may be written

$$C + U_1 = Q_v + U_2 \tag{4-48}$$

which gives

$$C = Q_v + (U_2 - U_1)_T \tag{4-49}$$

Strictly speaking, the heat value Q_v is an absolute value but varies with the temperature at which the test is conducted as indicated by the term $(U_2 - U_1)_T$.

In the constant-pressure calorimeter there is a steady flow of the air-fuel mixture into the calorimeter and a steady flow of combustion products out of it. Therefore the energy equation becomes

$$C + U_1 + APV_1 = Q_p + U_2 + APV_2 \tag{4-50}$$

or with expression (2-30)

$$C = Q_p + (H_2 - H_1)_T \tag{4-51}$$

Thus, similarly to Q_v, Q_p also varies with the temperature of the process.

As equations (4-49) and (4-51) show, chemical energy can be used only in connection with the tables of internal-energy values by means of which it was computed.

Heat values and chemical energies of reactions as encountered in combustion processes of engines are given in Table 4-3

Example 4-7.—Determine the value of chemical energy per mol of carbon monoxide, using as data its heat value Q_p given in Table 4-3.

The energy equation for this process is, by equation (4-50),

$$C_{\mathrm{CO}} + (U_{\mathrm{CO}} + 0.5U_{\mathrm{O}_2})_{537} + 1.5APV_1 = Q_p + (U_{\mathrm{CO}_2})_{537} + APV_2$$

Substituting values for U and $PV = RT$ from Table 2-2 and for Q_p from Table 4-3 gives

$$C_{\mathrm{CO}} + 81 + 0.5 \times 83 + 1.5 \times 1{,}066 = 121{,}721 + 115 + 1{,}066$$

or, solving for C_{CO},

$$C_{\mathrm{CO}} = 121{,}721 + 1{,}181 - 1722 = 121{,}180 \text{ Btu/mol}$$

which checks with the value given in Table 4-3.

TABLE 4-3.—HEAT VALUES AND CHEMICAL ENERGIES FOR GASEOUS FUELS

Fuel or substance	Symbol	Mol. wt per lb per mol	Reaction	Latent heat at 77 F, Btu per mol	High heat value at 77 F, Btu per mol		Chemical energy for the reaction indicated, for use only with Tables 2-2 and 4-2, Btu per mol C
					Q_p	Q_v	
Oxygen	O	16.0	$O = \frac{1}{2}O_2$	....	....	....	105,733
Hydroxide	OH	17.01	$OH = \frac{1}{2}(H_2 + O_2)$	....	....	....	10,700
Nitric oxide	NO	30.04	$NO = \frac{1}{2}(N_2 + O_2)$	....	....	....	38,746
Hydrogen	H	1.01	$H = \frac{1}{2}H_2$	....	....	....	92,822
Hydrogen	H_2	2.02	$H_2 + 0.5O_2 = H_2O$	....	122,963[a]	121,365	103,486
Carbon monoxide	CO	28.0	$CO + 0.5O_2 = CO_2$	....	121,721[b]	121,188	121,181
Methyl alcohol	CH_3OH	32.03	$CH_3OH + 1.5O_2 = CO_2 + 2H_2O$	16,092[c]	328,644[c]	327,045	291,374
Ethyl alcohol	C_2H_5OH	46.05	$C_2H_5OH + 3O_2 = 2CO_2 + 3H_2O$	18,216[e]	606,204[e]	604,072	550,553
Methane	CH_4	16.03	$CH_4 + 2O_2 = CO_2 + 2H_2O$	....	383,022[b]	380,891	345,214
Ethylene	C_2H_4	28.03	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	....	607,104[d]	604,973	569,298
Ethane	C_2H_6	30.05	$C_2H_6 + 3.5O_2 = 2CO_2 + 3H_2O$	....	671,058[e]	668,394	614,866
Benzene	C_6H_6	78.05	$C_6H_6 + 7.5O_2 = 6CO_2 + 3H_2O$	14,495[f]	1,417,041[g]	1,414,377	1,360,843
n-Heptane	C_7H_{16}	100.13	$C_7H_{16} + 11O_2 = 7CO_2 + 8H_2O$	15,750[h]	2,077,724[i]	2,072,394	1,929,633
n-Octane	C_8H_{18}	114.14	$C_8H_{18} + 12.5O_2 = 8CO_2 + 9H_2O$	17,730[h]	2,359,435[i]	2,353,572	2,192,964
n-Dodecane	$C_{12}H_{26}$	170.21	$C_{12}H_{26} + 18.5O_2 = 12CO_2 + 13H_2O$	23,000[j]	3,486,909[i]	3,478,914	3,248,917
Hexadecadiene	$C_{16}H_{30}$	222.24	$C_{16}H_{30} + 23.5O_2 = 16CO_2 + 15H_2O$	[30,000][k]	[4,370,000][k]	[4,363,000][k]	[4,075,000][k]

[a] *Natl. Bur. Standards, J. Research*, vol. 6, p. 1, 1931.
[b] *Ibid.*, vol. 6, p. 37, 1931.
[c] *Ibid.*, vol. 13, p. 189, 1934.
[d] *Ibid.*, vol. 13, p. 249, 1934.
[e] *Ibid.*, vol. 12, p. 735, 1934.
[f] *Ibid.*, vol. 6, p. 881, 1931.
[g] *Ibid.*, vol. 2, p. 375, 1929.
[h] *Ibid.*, vol. 13, p. 21, 1934.
[i] *Ibid.*, vol. 33, p. 225, 1944.
[j] *Natl. Bur. Standards, Misc. Pub.*, 1929, p. 97.
[k] Evaluated by extrapolation.

Chemical values are relative values because values from tables of internal energy and enthalpy, which depend upon the temperature, enter into their computation; therefore chemical values must be used only with these particular tables. If U and H were determined on an absolute basis, chemical-energy values would become absolute and would be equal to the heat value at absolute zero temperature.

Fuels containing hydrogen form H_2O as one of the products of combustion. Calorimetric determination of heat values is conducted under conditions which result in the condensation of practically all the water formed. Therefore the latent heat of vaporization of water, which at the usual calorimeter temperature of 77 F is equal to 1050 Btu/lb, must be taken into account.

Example 4-8.—Determine the chemical energy of 1 mol methane, CH_4, using as data its heat value Q_p given in Table 4-3.

The energy equation for this process is, by equation (4-50),

$$C_{CH_4} + (U_{CH_4} + 2U_{O_2})_{537} + 3APV_1$$
$$= Q_p + (U_{CO_2} + 2U_{H_2O})_{537} + 3APV_2 - 2M \times 1050$$

Taking the values for U from Table 2-2 and that for CH_4 from Example 2-3 and canceling $3APV_1 = 3APV_2$ gives

$$C_{CH_4} + (112 + 2 \times 83) = 383{,}022 + 115 + 2 \times 101 - 2 \times 18.016 \times 1050$$

or

$$C_{CH_4} = 383{,}022 + 317 - 37{,}897 - 278 = 345{,}214 \text{ Btu/mol}$$

Constant-pressure Combustion.—If equations (4-50) and (4-48) are compared, it is evident that

$$Q_p - Q_v = AP(V_1 - V_2) \qquad (4\text{-}52)$$

Since in calorimeteric tests the water vapor is condensed, the volume V_2 after combustion is taken without the volume of water.

Example 4-9.—Determine the difference between the heat values at constant-pressure and constant-volume combustion of 1 lb gasoline at atmospheric pressure and standard calorimeter temperature of 77 F.

Since the presence of nitrogen does not affect the process, combustion with theoretical air can be assumed, in which data from Table 4-2 can be used. This table gives

$$V_1 = 3.32 + 197.5 = 200.82 \text{ cu ft}$$

From a molecular equation, similar to equation (4-22), it is found that the number of mols before combustion of 1 mol of C_8H_{18} with theoretical air is

$$n_1 = 1 + 12.5(1 + 3.76) = 60.5 \text{ mols}$$

After combustion, with water condensed, the number of gaseous mols is

$$n_2' = 8 + 12.5 \times 3.76 = 55 \text{ mols}$$

Therefore the volume of products, at 60 F, is

$$V_2 = 200.82 \times 55/60.5 = 182.6 \text{ cu ft}$$

Thus, taking into account the temperature of 77 F, by equation (4-52),

$$Q_p - Q_v = 14.7 \times 144(200.82 - 182.6)537/(520 \times 778) = 51.2 \text{ Btu/lb}$$

From Table 4-2, $Q_v = 20{,}620$ Btu/lb, and the correction is only 0.25 per cent.

Combustion Temperatures.—The maximum temperature for the combustion process is attained if the combustion is at constant volume, complete, and adiabatic—without any heat losses and without energy supplied, other than of the reaction. The corresponding energy equation is

$$C + U_1 = U_2 \tag{4-53}$$

When the number of mols of the mixture and its temperature before combustion are known, the left side of equation (4-52) can be computed. With the number of mols of each constituent of the combustion products known, their temperature can be found by a cut-and-try method, using corresponding internal energies as given in Table 2-2.

If combustion is at constant pressure, the work of expansion $AP(V_2 - V_1)$ will be done, lowering U_2 and hence the final temperature. The energy equation becomes now

$$C + U_1 = U_2 + AP(V_2 - V_1) \tag{4-54}$$

or, using designation (2-30),

$$C + H_1 = H_2 \tag{4-55}$$

indicating that in this case the final temperature must be figured out using enthalpies instead of internal energies.

However, combustion reactions do not follow the simple equation (4-24). Because of dissociation at elevated temperatures, elements other than those given in equation (4-24) will appear in the products of combustion. Some of these additional constituents will contain unused chemical energy, and as a result the final temperatures will be considerably lower than those found from equation (4-53) or (4-55) in conjunction with equation (4-24). More accurate values of combustion temperatures are obtained from combustion charts, as explained later.

4-8. Chemical Equilibrium.[1]—Owing to dissociation when carbon and hydrogen react with oxygen of air, the reaction does not proceed to the point where all carbon and hydrogen are consumed, forming CO_2 and H_2O. Instead, the reactions proceed only until an equilibrium condition is reached in which not only the final products of the reaction, CO_2 and H_2O, are present, but also certain amounts of the original reaction substances and some intermediate compounds. The proportions, at equilibrium, of the different constituents of the products of combustion depend on the original proportions and on the temperature and pressure reached at the end of the reaction. The higher the final temperature, the less complete the combustion reaction. On the other hand, the higher the pressure, the more complete the reaction. In fuel-air mixtures at equilibrium after combustion, the following substances usually are present:

$$CO_2,\ H_2O,\ N_2,\ O_2,\ CO,\ H_2,\ OH,\ NO,\ H,\ O,\ C, \text{ and } CH_4$$

However, free carbon, C, and methane, CH_4, are present in such small portions that they can be disregarded in the following computations.

The determination of the proportions of the various substances at chemical equilibrium is based upon a relationship of the partial pressures. In general, for the reaction

$$aA + bB \rightleftarrows cC + dD \tag{4-56}$$

there exists the relationship

$$(P_C^c \times P_D^d)/(P_A^a \times P_B^b) = K \tag{4-57}$$

where a, b, c, and d are the respective numbers of mols of the substances A, B, C, and D, P is the partial pressure, in atmospheres, of the substance denoted by the subscript, and K is the equilibrium coefficient, which is a function of the temperature.

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[1] The method indicated follows in general the procedure used by R. L. Hershey, J. E. Eberhardt, and H. C. Hottel, Thermodynamic Properties of the Working Fluid in Internal-combustion Engines, *SAE Journal*, vol. 39, pp. 409*ff*., 1936.

Examples of such reactions, as encountered in combustion, are

$$CO_2 + H_2 \rightleftarrows CO + H_2O$$

for which, according to equation (4-57), the pressure relation is

$$(P_{CO})(P_{H_2O})/[(P_{CO_2})(P_{H_2})] = K_5$$

$$2H_2O \rightleftarrows 2H_2 + O_2$$

for which $$[(P_{H_2})^2 \times (P_{O_2})]/(P_{HO_2})^2 = K_3$$

The equilibrium coefficients for six possible reactions between the products of combustion have been determined for various temperatures and are given in Table 4-4. With these coefficients known, the partial pressure of each constituent can be computed from equations expressing ratios of certain pairs of constituents; in this way their relative amounts are found.

As a basis, P mols of products at a total pressure of P atmospheres are chosen. The partial pressure, or the number of mols of each constituent, is expressed by its chemical symbol in parentheses. If A is the number of mols of total O_2, in any form, per carbon atom in the system, then $3.76A$ is the number of mols of total N_2 per carbon atom, and with the designations of expression (4-24)

$$A = (1 + e)(n + 0.25m)/n \qquad (4\text{-}58)$$

If B is the number of mols of H_2 per carbon atom, then

$$B = m/2n \qquad (4\text{-}59)$$

With these designations the condition satisfying the known ratio of H_2 to N_2 in the equilibrium products is

$$(H_2O) + (H_2) + 0.5[(OH) + (H)] = [B/3.76A][(N_2) + 0.5(NO)] \qquad (4\text{-}60)$$

The condition satisfying the known ratio of O_2 to N_2 is

$$(O_2) + (CO_2) + 0.5[(O) + (OH) + (NO) + (CO) + (H_2O)] = [N_2 + 0.5(NO)]/3.76 \qquad (4\text{-}61)$$

The condition satisfying the known ratio of C to N_2 is

$$(CO_2) + (CO) = [(N_2) + 0.5(NO)]/3.76A \qquad (4\text{-}62)$$

Finally, the general condition that the total pressure P is equal to the sum of partial pressures gives the expression

$$P = (CO_2) + (H_2O) + (N_2) + (O_2) + (CO) + (H_2) + (OH) + (NO) + (O) + (H) \qquad (4\text{-}63)$$

There are six combustion reactions, (1) to (6), Table 4-4, from

TABLE 4-4.—EQUILIBRIUM CONSTANTS IN COMBUSTION REACTIONS, EQUATION (4-56)
(Atmospheres)

Temperature, °R	(1) $\frac{1}{2}H_2 = H$ K_1	(2) $\frac{1}{2}O_2 = O$ K_2	(3) $2H_2O = 2H_2 + O_2$ K_3	(4) $H_2O = OH + \frac{1}{2}H_2$ K_4	(5) $CO_2 + H_2 = CO + H_2O$ K_5	(6) $H_2O + \frac{1}{2}N_2 = H_2 + NO$ K_6
3000	1.13×10^{-4}	0.343×10^{-4}	0.191×10^{-9}	0.148×10^{-4}	3.311	0.978×10^{-7}
3100	1.91×10^{-4}	0.624×10^{-4}	0.617×10^{-9}	0.282×10^{-4}	3.503	2.14×10^{-7}
3200	3.13×10^{-4}	1.085×10^{-4}	1.906×10^{-9}	0.513×10^{-4}	3.715	4.52×10^{-7}
3300	4.96×10^{-4}	1.820×10^{-4}	5.49×10^{-9}	0.891×10^{-4}	3.917	9.23×10^{-7}
3400	7.68×10^{-4}	2.952×10^{-4}	14.8×10^{-9}	1.514×10^{-4}	4.130	17.99×10^{-7}
3500	1.16×10^{-3}	0.462×10^{-3}	0.380×10^{-7}	0.248×10^{-3}	4.345	0.339×10^{-5}
3600	1.72×10^{-3}	0.708×10^{-3}	0.912×10^{-7}	0.398×10^{-3}	4.550	0.610×10^{-5}
3700	2.49×10^{-3}	1.072×10^{-3}	2.09×10^{-7}	0.624×10^{-3}	4.742	1.059×10^{-5}
3800	3.51×10^{-3}	1.59×10^{-3}	4.57×10^{-7}	0.945×10^{-3}	4.932	1.797×10^{-5}
3900	4.90×10^{-3}	2.32×10^{-3}	9.77×10^{-7}	1.412×10^{-3}	5.117	3.021×10^{-5}
4000	6.76×10^{-3}	3.31×10^{-3}	0.200×10^{-5}	2.07×10^{-3}	5.285	0.490×10^{-4}
4100	9.12×10^{-3}	4.65×10^{-3}	0.389×10^{-5}	3.02×10^{-3}	5.445	0.767×10^{-4}
4200	0.0122	6.46×10^{-3}	0.741×10^{-5}	4.32×10^{-3}	5.598	1.188×10^{-4}
4300	0.0159	8.82×10^{-3}	1.35×10^{-5}	6.03×10^{-3}	5.754	1.80×10^{-4}
4400	0.0204	11.75×10^{-3}	2.40×10^{-5}	8.23×10^{-3}	5.888	2.66×10^{-4}
4500	0.0260	0.01549	0.407×10^{-4}	0.0111	6.012	0.385×10^{-3}
4600	0.0328	0.0202	0.676×10^{-4}	0.0146	6.124	0.544×10^{-3}
4700	0.0412	0.0260	1.122×10^{-4}	0.0193	6.237	0.767×10^{-3}
4800	0.0513	0.0331	1.862×10^{-4}	0.0251	6.353	1.08×10^{-3}
4900	0.0631	0.0417	3.021×10^{-4}	0.0328	6.457	1.50×10^{-3}
5000	0.0777	0.0525	0.468×10^{-3}	0.0417	6.546	2.02×10^{-3}
5100	0.0945	0.0653	0.708×10^{-3}	0.0525	6.653	2.66×10^{-3}
5200	0.1148	0.0804	1.047×10^{-3}	0.0733	6.745	3.47×10^{-3}
5300	0.1381	0.0978	1.585×10^{-3}	0.0813	6.839	4.57×10^{-3}
5400	0.1622	0.1178	2.396×10^{-3}	0.1002	6.919	6.06×10^{-3}

which six equilibrium relations can be computed. These six relations together with the above four equations, (4-60) to (4-63), permit solving for the ten components.

The equilibrium relations are

$$(\mathrm{H}) = K_1 \sqrt{(\mathrm{H}_2)} \tag{4-64}$$
$$(\mathrm{O}) = K_2 \sqrt{(\mathrm{O}_2)} \tag{4-65}$$
$$(\mathrm{O}_2) = K_3[(\mathrm{H}_2\mathrm{O})/(\mathrm{H}_2)]^2 \tag{4-66}$$
$$(\mathrm{OH}) = K_4(\mathrm{H}_2\mathrm{O})/\sqrt{(\mathrm{H}_2)} \tag{4-67}$$
$$(\mathrm{CO}_2) = (\mathrm{CO})(\mathrm{H}_2\mathrm{O})/[K_5(\mathrm{H}_2)] \tag{4-68}$$
$$(\mathrm{NO}) = K_6(\mathrm{H}_2\mathrm{O})\sqrt{(\mathrm{N}_2)}/(\mathrm{H}_2) \tag{4-69}$$

Substituting from equations (4-67), (4-64), and (4-69) in equation (4-60), and solving for (H_2O) give

$$\mathrm{H}_2\mathrm{O} = \frac{B(\mathrm{N}_2)/3.76A - (\mathrm{H}_2) - 0.5K_1\sqrt{(\mathrm{H})_2}}{1 + 0.5K_4/\sqrt{(\mathrm{N}_2)} - 0.5\mathrm{K}_6B\sqrt{(\mathrm{N}_2)}/[3.76A(\mathrm{H}_2)]} \tag{4-70}$$

Substituting for (CO_2) its expression (4-68) in equation (4-61) and solving for (CO) give

$$(\mathrm{CO}) = \frac{K_5(\mathrm{H}_2)[(\mathrm{N}_2 + 0.5(\mathrm{NO})]}{3.76A[(\mathrm{H}_2\mathrm{O}) + K_5(\mathrm{H}_2)]} \tag{4-71}$$

The procedure of solving the equations for fixed values of A and B and temperature is one of trial and error. If the total pressure for which the calculations are conducted is P, the value of (N_2) can be estimated; depending upon air-fuel ratios and temperatures, it varies only from $0.70P$ to $0.75P$. Then at the desired temperature, which fixes the various K values, Table 4-4, a value for (H_2) is tentatively assumed; the higher the air-fuel ratio, the lower will be this value. Then in the order given are calculated (H_2O) from equation (4-70), (NO) from equation (4-69), (CO) from equation (4-71), and (O_2) from equation (4-66); the others, (H), (O), (OH), and (CO_2), are computed in any order from equations (4-64) to (4-68). Then the assumed value of (H_2) is tested by substituting it in the oxygen balance, equation (4-61). If the left side is greater than the right one, (H_2) was assumed too small, and vice versa. A new value for (H_2) is assumed, and the calculations are repeated until equation (4-61) is satisfied. Finally the total pressure of the gas mixture is found in atmospheres from equation (4-63) and by multiplying by 14.7 is converted into pounds per square inch absolute.

With the composition (mols) known at a given temperature and pressure, the values of volume V, internal energy U, chemical energy C, enthalpy H, and entropy S can be determined for the equilibrium mixture.

The entropy S of each constituent can be calculated by equation (2-40). To facilitate these computations the entropies per mol of the 10 constituents were calculated in advance for different temperatures and a pressure of 1 atm and presented in Table 4-5. The entropies of CO_2, H_2O, N_2, and O_2 were arbitrarily assumed as equal to 0 Btu/lb at 520 R. The entropies of the other six constituents at 520° are equal to the entropy of formation of these gases from the first four.

Example 4-10.—Determine the values of P, V, U, C, and S at 5000 R for the reaction between $C_{16}H_{30}$ and 25 per cent excess air; the pressure of the equilibrium mixture to be about 600 psia.

According to a preliminary trial, nitrogen will be assumed to be

$$0.734(600/14.7) = 29.95 \text{ mols}$$

By equation (4-58)

$$A = (1 + 0.25)(16 + 0.25 \times 30)/16 = 1.836$$

and

$$3.76A = 3.76 \times 1.836 = 6.903$$

By equation (4-59)

$$B = 30/(2 \times 16) = 0.9375$$

From a preliminary trial assume $(H_2) = 0.0683$ mol.

Using these values for (N_2), (H_2), A, and B, and taking the values for the equilibrium constants from Table 4-4 for the temperature of 5000 F, the values for the 10 constituents were computed by means of equations (4-64) to (4-71) in the order indicated above. The computed values are given in the second column of Table 4-6. A check by equation (4-61) gives for the left side 8.0431 mols, for the right side 8.0457, a discrepancy of 0.033 per cent, hence negligible. A second check by equation (4-63) gives

$$P = 600/14.7 = 40.816 \text{ atm}$$

whereas Table 4-6 shows the sum of partial pressures (mols) as 40.815, an almost perfect agreement.

In column (3), Table 4-6, are given internal energies taken from Table 2-2 for 5000 R. Column (4) gives the energy of the amount present, and in column (5) are given data of C from Table 4-3. It should be observed that the total chemical energy of H is equal to the sum of the energies of the two reactions $H = \frac{1}{2}H_2$ and $\frac{1}{2}(H_2 + 0.5O_2) = \frac{1}{2}(H_2O)$, Table 4-3, or

$$C_H = 92{,}822 + 0.5 \times 103{,}486 = 144{,}565 \text{ Btu}$$

Similarly, for OH the chemical energy C_{OH} is also equal to the sum of two reactions:

$$C_{OH} = 10{,}700 + 0.5 \times 103{,}486 = 62{,}443 \text{ Btu}$$

TABLE 4-5.—ENTROPY OF COMBUSTION GASES AT 14.7 PSI[a]
(Btu per mol per °R)

Temp., °R	CO_2	H_2O	N_2	O_2	CO	H_2	OH	NO	O	H
520	0	0	0	0	20.70	10.54	9.07	2.97	13.52	15.83
600	1.30	1.22	1.00	1.03	21.71	11.55				
700	2.76	2.43	2.08	2.16	22.79	12.64				
800	4.10	3.48	3.01	3.12	23.73	13.55				
900	5.34	4.46	3.84	3.98	24.57	14.37				
1000	6.50	5.36	4.59	4.76	25.33	15.11				
1100	7.57	6.20	5.27	5.48	26.02	15.78				
1200	8.57	6.97	5.91	6.16	26.66	16.39				
1300	9.51	7.67	6.50	6.80	27.27	16.95				
1400	10.41	8.34	7.05	7.39	27.82	17.47				
1500	11.26	8.97	7.56	7.94	28.31	17.96				
1600	12.06	9.57	8.05	8.47	28.79	18.43				
1700	12.83	10.15	8.52	8.97	29.27	18.87				
1800	13.57	10.71	8.97	9.44	29.75	19.29				
1900	14.28	11.23	9.40	9.89	30.19	19.69				
2000	14.95	11.75	9.81	10.32	30.60	20.06				
2100	15.59	12.25	10.20	10.73	31.00	20.41				
2200	16.21	12.75	10.57	11.12	31.38	20.74				
2300	16.81	13.22	10.92	11.50	31.74	21.07				
2400	17.39	13.67	11.26	11.86	32.09	21.39				
2500	17.95	14.11	11.60	12.21	32.43	21.69				
2600	18.49	14.53	11.93	12.55	32.76	21.99				
2700	19.02	14.94	12.24	12.88	33.07	22.29				
2800	19.53	15.35	12.54	13.21	33.37	22.57				
2900	20.03	15.74	12.83	13.53	33.66	22.83				
3000	20.51	16.13	13.11	13.84	33.95	23.09	21.81	16.50	22.22	24.53
3100	20.98	16.51	13.39	14.14	34.23	23.35	22.08	16.79	22.39	24.69
3200	21.44	16.88	13.67	14.43	34.51	23.61	22.34	17.07	22.54	24.85
3300	21.89	17.25	13.94	14.70	34.78	23.87	22.59	17.34	22.69	25.00
3400	22.32	17.61	14.20	14.96	35.04	24.12	22.84	17.61	22.84	25.15
3500	22.74	17.96	14.45	15.21	35.29	24.36	23.08	17.87	22.99	25.30
3600	23.15	18.30	14.69	15.46	35.54	24.58	23.31	18.11	23.13	25.44
3700	23.54	18.64	14.92	15.71	35.77	24.80	23.53	18.34	23.26	25.57
3800	23.91	18.97	15.15	15.96	35.99	25.02	23.75	18.57	23.39	25.70
3900	24.28	19.29	15.37	16.20	36.21	25.23	23.97	18.79	23.51	25.82
4000	24.65	19.61	15.58	16.43	36.43	25.44	24.19	19.02	23.63	25.94
4100	25.02	19.92	15.79	16.65	36.65	25.64	24.40	19.24	23.75	26.06
4200	25.38	20.22	16.00	16.87	36.87	25.84	24.61	19.46	23.87	26.18
4300	25.73	20.51	16.21	17.08	37.08	26.04	24.81	19.68	23.99	26.30
4400	26.08	20.79	16.42	17.29	37.29	26.24	25.01	19.89	24.11	26.42
4500	26.42	21.07	16.62	17.50	37.49	26.44	25.20	20.09	24.23	26.54
4600	26.74	21.35	16.81	17.70	37.69	26.63	25.39	20.28	24.34	26.65
4700	27.06	21.62	17.00	17.90	37.88	26.82	25.58	20.47	24.45	26.76
4800	27.37	21.89	17.18	18.10	38.06	27.00	25.76	20.66	24.56	26.87
4900	27.68	22.16	17.36	18.30	38.24	27.18	25.94	20.85	24.67	26.98
5000	27.98	22.42	17.54	18.49	38.42	27.35	26.12	21.03	24.78	27.09
5100	28.28	22.68	17.72	18.68	38.59	27.52	26.29	21.21	24.88	27.19
5200	28.57	22.93	17.89	18.86	38.76	27.69	26.46	21.38	24.97	27.28
5300	28.86	23.18	18.06	19.04	38.93	27.86	26.63	21.55	25.06	27.37
5400	29.14	23.42	18.23	19.22	39.10	28.02	26.79	21.72	25.14	27.45

TABLE 4-6.—CALCULATION OF INTERNAL AND CHEMICAL ENERGIES

(1)	(2)	(3)	(4)	(5)	(6)
Constituent gas	Quantity, mols	Internal energy		Chemical energy	
		Btu/mol	Btu	Btu/mol	Btu
N_2	29.9500	27,589	826,300		
H_2	0.0683	25,819	1,763	103,486	7,060
H_2O	3.7321	39,885	148,800		
NO	0.6040	28,570	17,250	38,746	23,400
CO	0.4688	27,907	13,080	121,181	56,800
O_2	1.3994	29,616	41,400		
H	0.0203	13,346	271	144,565	2,935
O	0.0621	13,346	828	105,733	6,570
OH	0.5956	26,319	15,670	62,443	37,180
CO_2	3.9144	51,365	201,000		
Sum...	40.8150	......	1,266,362	....	133,945

TABLE 4-7.—CALCULATION OF ENTROPY

(1)	(2)	(3)	(4)	(5)	(6)	(7)
Constituent gas	Mols M and pressure P	Entropy at 1 atm		$RM = 1.986M$	$\log_e M = \log_e P$	$RM \times \log_e P$
		Btu/mol-°R	Btu/°R			
N_2	29.9500	17.54	525.32	59.565	3.3996	202.52
H_2	0.0683	27.35	1.87	0.136	−2.6839	−0.36
H_2O	3.7321	22.42	83.60	7.420	1.3170	9.76
NO	0.6040	21.03	13.35	1.200	−0.6042	−0.72
CO	0.4688	38.42	18.03	0.932	−0.7576	−0.71
O_2	1.3994	18.49	25.88	2.779	0.3361	0.93
H	0.0203	27.09	0.55	0.040	−3.9572	−0.16
O	0.0621	24.78	1.54	0.123	−2.7790	−0.34
OH	0.5956	26.12	15.55	1.182	−0.5182	−0.61
CO_2	3.9144	27.98	109.52	7.780	1.3646	10.61
Sum...	40.8150	.....	795.21	......	....	220.92

From the characteristic equation (2-6) for M mols

$$V = \frac{MRT}{P} = \frac{40.815 \times 1545 \times 5000}{600 \times 144} = 3649 \text{ cu ft}$$

The entropy was computed according to equation (2-40). The first terms in that equation, or entropy at 1 atm, were taken from Table 4-5 and are presented in column (3) of Table 4-7. Columns (5), (6), and (7) present calculations necessary to determine the second term in equation (2-40).

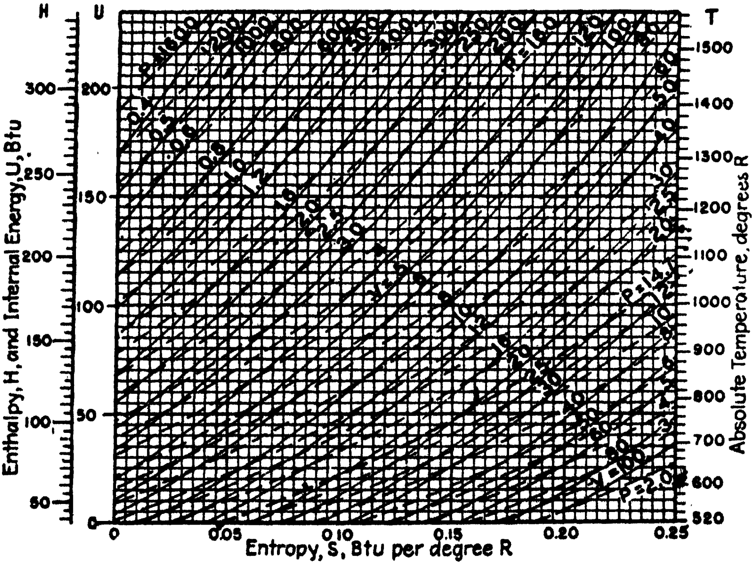

FIG. 4-2.—Compression chart for a correct fuel-air mixture, 100 per cent theoretical air. P = pressure, psia, broken lines; V = volume, cu ft, solid lines.

Fuel octane, C_8H_{18}, weight = 0.05985 lb; air weight = 0.90 lb; residual combustion products f = 0.10 lb; fuel-air ratio = 0.0665; n_1 = lb-mols of charge for compression = $0.0353 + 0.002f$; energy of combustion $C = 1280(1 - f)$; H of combustion $= 1278(1 - f)$.

The entropy for the full number of mols, by equation (2-40), is

$$S = S_{1\text{atm}} - \Sigma MR \log_e P = 795.21 - 220.92 = 574.29 \text{ Btu/°R}$$

Total N_2 = N_2 + 0.5 NO = 29.950 + 0.5 × 0.604 = 30.252 mols
Total O_2 = 30.252/3.76 = 8.045 mols
Weight of air = (30.252 + 8.045)28.95 = 1108.6 lb

Dividing all the results by 1108.6 gives values for a mixture which originally contained 1 lb of air:

T = 5000 R; P = 600 psi; U = 1,142.3 Btu/lb; $U + C$ = 1,273.7 Btu/lb;
v = 3.292 cu ft/lb; C = 131.4 Btu/lb; S = 0.5185 Btu/lb-°R.

4-9. Combustion Charts.—Applying the procedure illustrated by Example 4-10 to a number of temperatures and pressures and presenting the obtained data in a graphical form, using entropies for abscissas and internal energies and enthalpies for ordinates, charts similar to Mollier steam charts can be constructed and used for solving various combustion problems. Unfortunately separate charts must be constructed for different fuels and different air-fuel ratios.

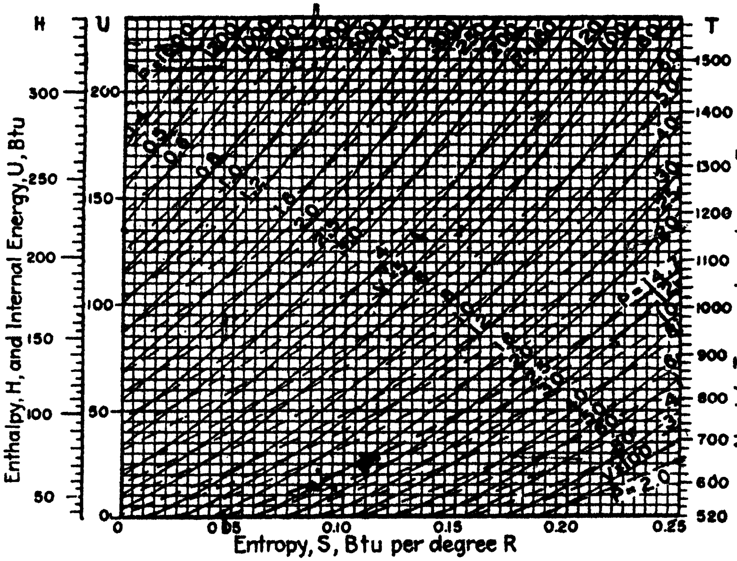

Fig. 4-3.—Compression chart for a rich fuel-air mixture, 85 per cent theoretical air. P = pressure, psia, broken lines; V = volume, cu ft, solid lines.

Fuel octane, C_8H_{18}, weight = 0.07038 lb; air weight = 0.90 lb; residual combustion product f = 0.10 lb; fuel-air ratio = 0.0782; n_1 = lb-mols of charge for compression = $0.0354 + 0.004f$; energy of combustion $C = 1507(1 - f) + 300f$; H of combustion $= 1504(1 - f) + 300f$.

Charts for the air-fuel mixture before combustion can be constructed in a similar way and used for the compression stroke. Such charts for gasoline (C_8H_{18}) for three typical air-fuel ratios are presented in Figs. 4-2 to 4-4.[1]

Combustion charts for the same conditions are presented in Figs. 4-5 to 4-7.[1] All six charts are constructed for 1 lb air and the corresponding amount of fuel.

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[1] Using as a basis Hershey, Eberhardt, and Hottel, *loc. cit.*

Figure 4-8 presents a chart of thermodynamic properties of 1 lb atmospheric air having an average or *standard* humidity, 0.991 lb air and 0.009 lb water.[1] This chart can be used with sufficient accuracy for the compression in a compression-ignition engine.

Figure 4-9 represents a combustion chart for 1 lb gas mixture representing the composition of the charge in a compression-ignition oil engine operating on $C_{16}H_{30}$ as fuel, with 50 per cent excess air and a 15:1 compression ratio. This mixture consists of 0.91345 lb air, 0.00826 lb water corresponding to standard humidity, 0.04193 lb fuel, and 0.03636 lb residual products of combustion.[2] Until additional charts are available, Fig. 4-9 may be used for other excess-air and compression-ratio conditions as found at full load in present engines.

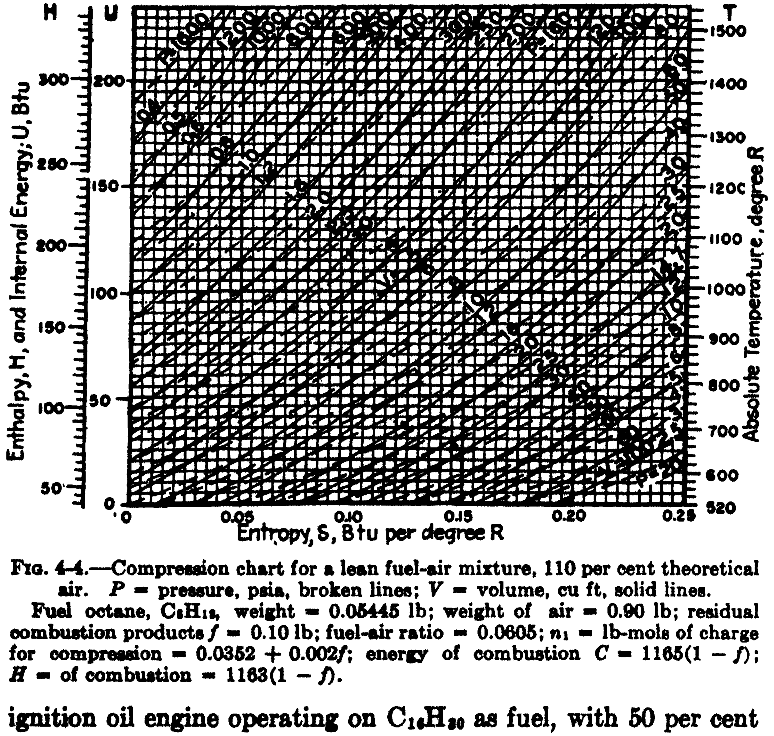

FIG. 4-4.—Compression chart for a lean fuel-air mixture, 110 per cent theoretical air. P = pressure, psia, broken lines; V = volume, cu ft, solid lines. Fuel octane, C_8H_{18}, weight = 0.05445 lb; weight of air = 0.90 lb; residual combustion products f = 0.10 lb; fuel-air ratio = 0.0605; n_1 = lb-mols of charge for compression = $0.0352 + 0.002f$; energy of combustion $C = 1165(1 - f)$; H = of combustion = $1163(1 - f)$.

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[1] See also MCGREGOR, J. R., The Influence of Humidity on Knock Rating, *SAE Journal*, vol. 40, p. 249, 1937.

[2] MALEEV, V. L., Combustion Charts for Compression Ignition Oil Engines, a paper to be published at a later date.

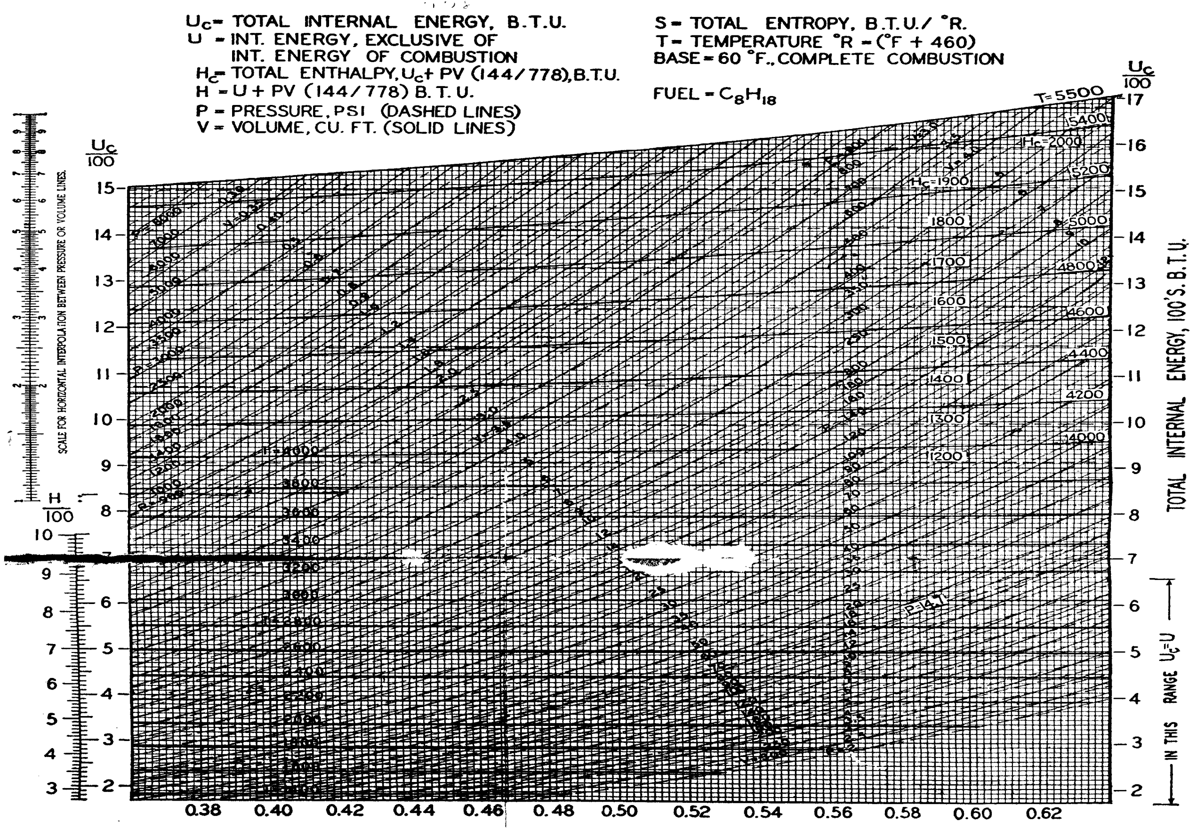

FIG. 4-5.—Combustion chart for an octane-air mixture, 100 per cent theoretical air. (*Hershey, Eberhardt, and Hottel.*)

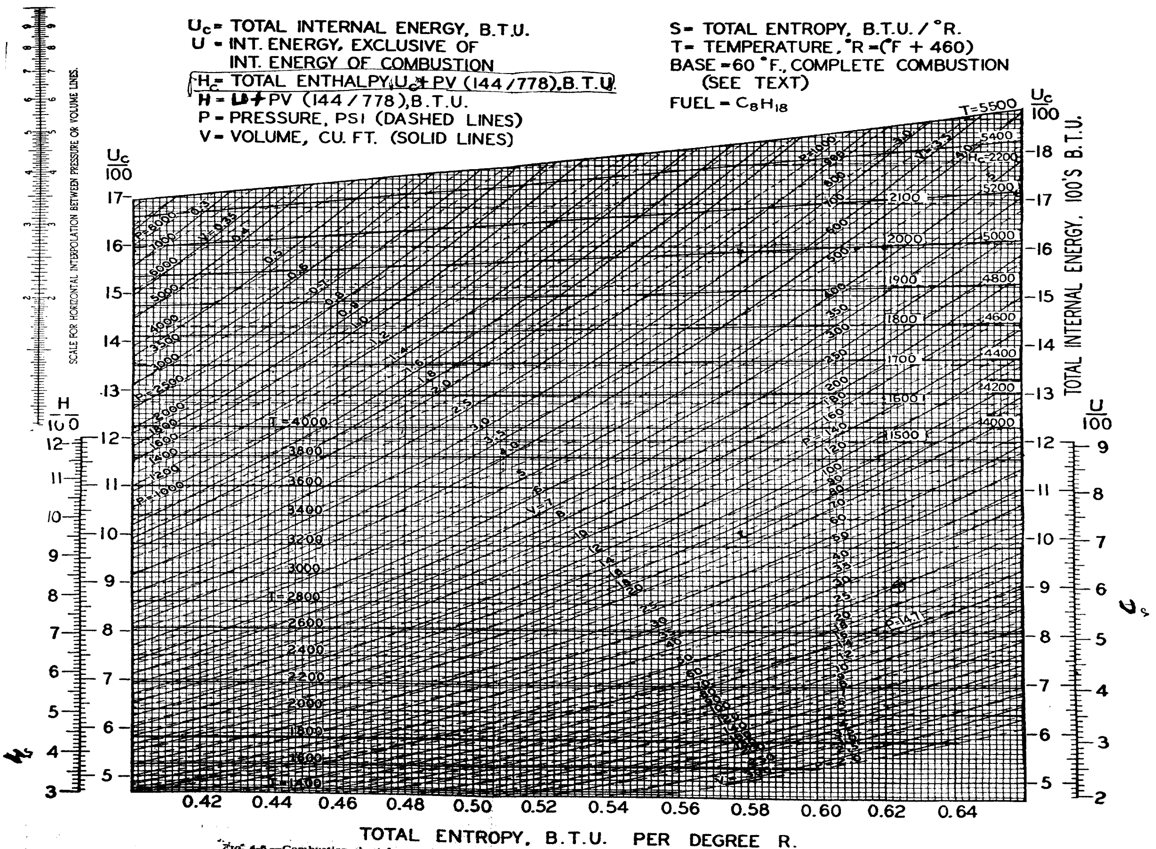

FIG. 4-6.—Combustion chart for an octane-air mixture, 85 per cent theoretical air. (*Hershey, Eberhardt, and Hottel.*)

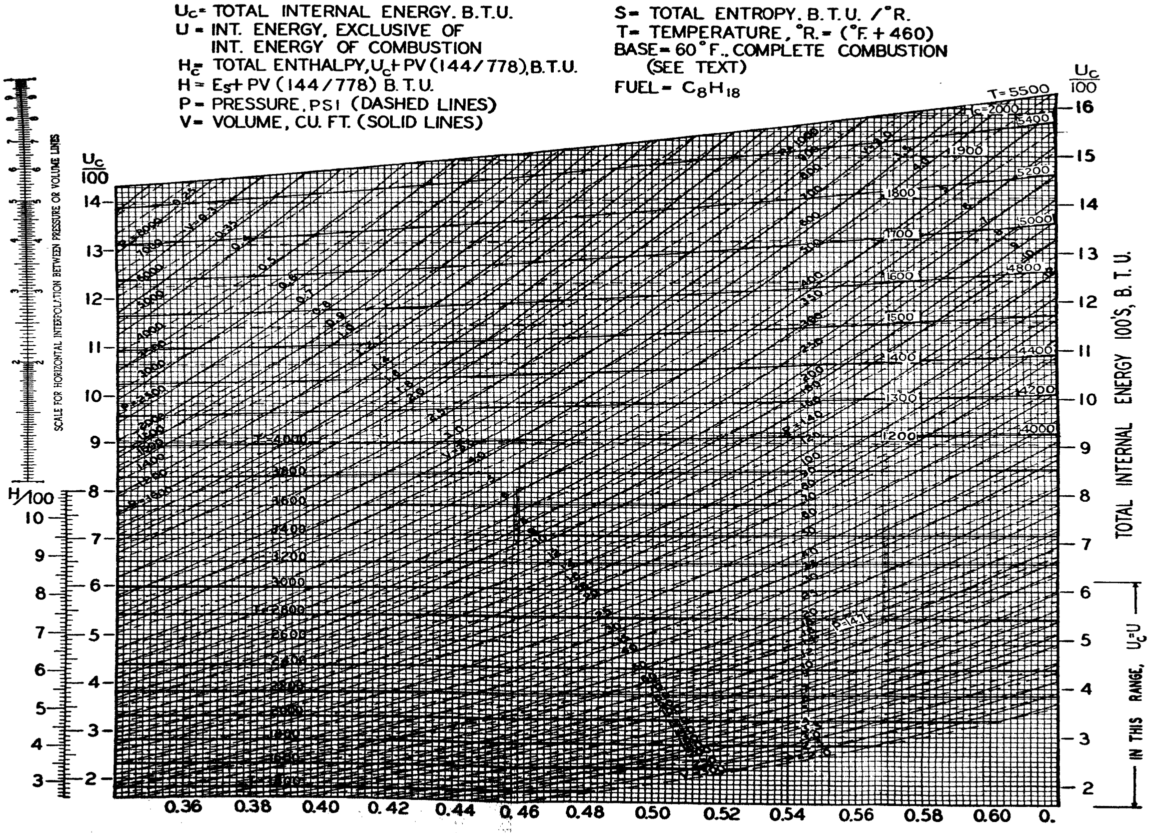

Fig. 4-7.—Combustion chart for an octane-air mixture, 110 per cent theoretical air. (*Hershey, Eberhardt, and Hottel.*)

If it is desired to compare Fig. 4-9 with Figs. 4-5 to 4-7, *i.e.*, to use as a basis not 1 lb gas mixture but 1 lb dry air with combustion products, both the ordinates and abscissas of Fig. 4-9 must be multiplied by 1.054.

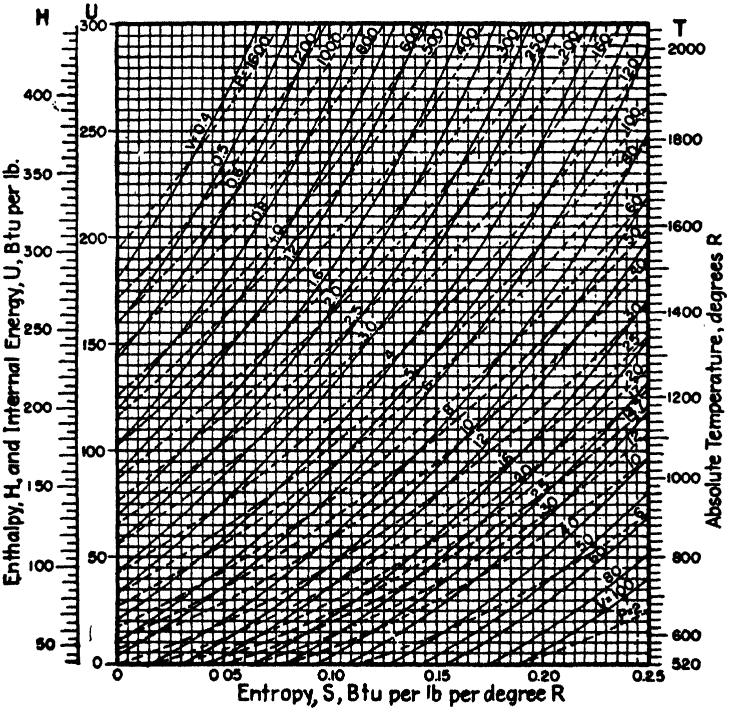

FIG. 4-8.—Compression chart for air. n_1 = lb-mols of air = 0.03364. P = pressure, psia, broken lines; V = volume, cu ft, solid lines.

4-10. Problems.—**1.** Deduce the combustion equations for: (*a*) complete combustion of 1 cu ft ethylene, C_2H_4, in air; (*b*) combustion with 25 per cent excess air.

2. Deduce the combustion equations for: (*a*) complete combustion in theoretical air of 1 mol and 1 lb octane, C_8H_{18}; (*b*) combustion with 12 per cent excess air.

3. Deduce the combustion equations for: (*a*) complete combustion of 1 lb engine fuel oil, $C_{16}H_{30}$, with theoretical air; (*b*) combustion with 25 per cent excess air; (*c*) combustion with 45 per cent excess air.

4. Find the amount of air necessary for combustion 1 lb octane, C_8H_{18}, with theoretical air.

5. Find the amount of air necessary for combustion of 1 lb fuel oil, $C_{16}H_{34}$, with (*a*) 25 per cent excess air and (*b*) 45 per cent excess air.

6. Find the amount of air necessary for combustion with 25 per cent excess air of 100 cu ft natural gas which has an analysis of 2.5 per cent H_2, 0.5 per cent CO, 92.0 per cent CH_4, 1.0 per cent O_2, 0.2 per cent CO_2, and 3.8 per cent N_2.

7. Find the volume of the combustion products formed by burning 1 lb carbon to CO_2 with 60 per cent excess air: (*a*) at a standard pressure and temperature and (*b*) at a pressure of 0.3 psi above atmospheric and a temperature of 720 F.

8. Find the volume of the combustion products of 1 cu ft methane with 32 per cent excess air at a pressure 0.4 psi above atmospheric and a temperature of 1000 F.

9. Find the volume of the combustion products of 1 lb fuel oil, $C_{16}H_{30}$, with 40 per cent excess air at a pressure of 0.45 psi above atmospheric and a temperature of 600 F.

10. Find the volume of the combustion products of 25 cu ft propylene with 48 per cent excess air, reduced to standard conditions.

11. Find the weight of the combustion products of 15 lb kerosene (dodecane, $C_{12}H_{26}$) with 52 per cent excess air.

12. Find the weight of the combustion products of 8 lb light kerosene (decane, $C_{10}H_{22}$) with 45 per cent excess air.

13. Find the weight of the combustion products of 7 lb gasoline, C_8H_{18}, with 10 per cent excess air.

14. Find the weight of the combustion products of 12 lb diesel fuel oil, $C_{16}H_{30}$, with 30 per cent excess air.

15. Find the weight of the combustion products with the data of problem 6.

16. A mixture of 80 per cent grain alcohol and 20 per cent wood alcohol is used as fuel. Find for 1 lb of this fuel: (*a*) the amount of air necessary for a complete combustion, (*b*) the volume at the combustion products at a pressure of 15 psia and a temperature of 575 F.

17. A mixture of 90 per cent octane with 10 per cent ethyl alcohol is used as fuel. With 10 per cent excess air find for 1 lb of this fuel: (*a*) the amount of air necessary; (*b*) the volume of the combustion products at a pressure of 15.1 psia and a temperature of 1000 F.

18. Find the air-fuel ratio and excess of air present when the exhaust-gas analysis showed CO_2, 12.5 per cent; O_2, 2.9 per cent; and N_2, 84.5 per cent. The analysis of the fuel gave 84.0 per cent carbon and 15.7 per cent hydrogen.

19. Find the air-fuel ratio and excess of air present when the exhaust-gas analysis by an Orsat apparatus showed 11.1 per cent CO_2; 5.7 per cent O_2; and the balance N_2. The fuel used was fuel oil $C_{16}H_{30}$.

20. Deduce a general expression for the volume change through combustion of a hydrocarbon fuel burned with a theoretical amount of air; equation (4-37) may be helpful to start.

21. Deduce a general expression for the volume change through combustion of a hydrocarbon fuel burned with an excess of air.

22. Find the partial pressures of the combustion products formed by gasoline, C_8H_{18}, burned with 5 per cent excess air: (*a*) at 1000 F and (*b*) at 70 F.

23. Find the partial pressures of the combustion products formed by kerosene, $C_{12}H_{26}$, burned with 45 per cent excess air: (*a*) at 900 F and (*b*) at 60 F.

24. Find the partial pressures of the combustion products formed by compression-ignition engine fuel oil, $C_{16}H_{30}$, burned with 38 per cent excess air: (*a*) at 650 F and (*b*) at 100 F.

25. Find the gas constant of the combustion products of problem 22 with all water vapor present and also the ratio of the gas constants after and before combustion.

26. Work problem 25 using data from problem 23.

27. Work problem 25 using data from problem 24.

28. Determine the imaginary and actual volumes at 15 psia of combustion products of 1 lb fuel using data from problem 22.

29. Work problem 28 using data from problem 23.

30. Work problem 28 using data from problem 24.

31. Compute the higher and lower heat values per cubic foot of natural gas as given in problem 6.

32. Compute the higher and lower heat values per cubic foot of butane and compare them with the values given in Table 4-2.

33. Compute the higher and lower heat values per pound of *n*-octane and compare them with the values given in Table 4-2.

34. Compute the higher and lower heat values per pound of a gasoline that has a specific gravity of 0.745.

35. Compute the higher and lower heat values per pound of kerosene that has a specific gravity of 0.822.

36. Compute the heat values per pound of fuel oil that has a specific gravity of 0.840.

37. Determine the chemical energy per mol of hydrogen: (*a*) using as data its heat value at constant volume, Table 4-3; (*b*) using as data its heat value at constant pressure.

38. Determine the chemical energy of 1 lb *n*-octane and check it with the value given in Table 4-3.

39. Determine the difference between the heat values of constant-pressure and constant-volume combustion of 1 lb dodecane and find its percentage referred to the total heat value based on Table 4-3.

40. Determine the difference between the heat values of constant-pressure and constant-volume combustion of *n*-heptane in percentage of the latter and check with Table 4-3.

41. Compute the maximum temperature developed by combustion of gasoline, C_8H_{18}, at a constant volume, without any heat loss, with theoretical air; initial conditions $t_1 = 700$ F, $p = 150$ psia. Compare it with the temperature as found by means of combustion chart, Fig. 4-5.

42. Determine the equilibrium composition and the volume of a mixture of 1 lb air with the theoretically correct amount of gasoline, C_8H_{18}, at a temperature of 3600 F and a pressure of 300 psia; check the answers with the corresponding combustion chart.

43. Using the composition as computed in problem 42 find the internal and chemical energies and the entropy of the equilibrium mixture defined by data of problem 42; check the answers with the corresponding combustion chart.

44. Work problems 42 and 43 for a mixture with 10 per cent excess air.

45. Determine the equilibrium composition and the volume of a mixture of 1 lb of air with such an amount of fuel oil, $C_{16}H_{30}$, as to give 50 per cent excess air. The final conditions are 3600 F and 900 psia.

46. Using the composition as computed in problem 45 find the internal and chemical energies and the entropy of the equilibrium mixture defined by data of problem 45; check the answers with the corresponding combustion chart.

CHAPTER 5

THEORETICAL ENGINE CYCLES

5-1. Working Cycles.—The operation of an internal-combustion engine goes through a cycle. The term is used, even though the combustion process is not reversible and the working substance does not go through a cycle. Many cycles have been proposed for internal-combustion engines; many have been tried, but so far only three have proved practical. The main difference in these cycles is in the combustion.

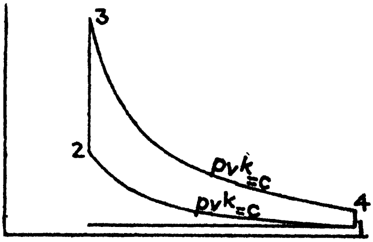

FIG. 5-1.—Explosion cycle.

The majority of modern engines operate on a cycle with combustion at constant volume. Theoretically this cycle is composed of an adiabatic compression line 1-2, Fig. 5-1, a constant volume combustion line 2-3, when heat is added, an adiabatic expansion line 3-4, and a constant-volume line 4-1 of heat rejection, bringing the operation back to the starting point 1. This cycle is often called the *explosion cycle* on account of the theoretically instantaneous combustion giving a sudden pressure increase. Sometimes this cycle is also called the *Otto cycle.*

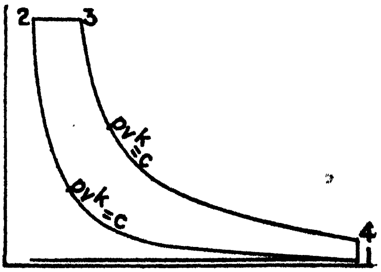

FIG. 5-2.—Combustion cycle.

Compression-ignition air-injection engines use a cycle with combustion at constant pressure, called a *combustion cycle* or a *Diesel cycle.* In this cycle the compression line 1-2, Fig. 5-2, is also an adiabatic; combustion occurs at constant pressure, line 2-3; the expansion line is an adiabatic; and heat is rejected along a line of constant volume 4-1.

A third cycle is obtained with part of the combustion occurring at constant volume, line 2-3, Fig. 5-3, and the rest of the combustion at constant pressure, 3-4; with adiabatic lines of compression, 1-2, expansion, 4-5, and a constant-volume line 5-1 of heat rejection. This cycle is known as a *combination* or *dual-combustion cycle.*

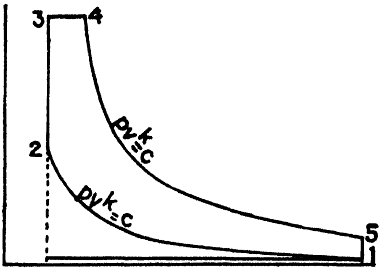

FIG. 5-3.—Combination cycle.

Compression Ratio.—A very important characteristic of an internal-combustion engine is the so-called *compression ratio* designated as r. Compression ratio is the volume V_1 of the gases at the beginning of the compression stroke, Fig. 5-1, 5-2, or 5-3, divided by the volume V_2 of the gases at the end of the compression stroke

$$r = V_1/V_2 \qquad (5\text{-}1)$$

The volume V_2 is called the *compression space* of an engine. The *piston displacement* of an engine is equal to the difference of the volumes at the two dead centers, $V_1 - V_2$.

5-2. Air-standard Cycle.—An accurate analysis of internal-combustion-engine cycles is a difficult problem, because of the chemical reactions that take place and because of the heat interchanges between the working gases and the cylinder walls enclosing them. To simplify the analysis the following conditions are set for the theoretical cycles:

1. The heat developed during combustion is considered simply as heat added from an outside source.
2. Likewise, the heat carried away with the escaping exhaust gases is considered as though it were taken away from the working substance, which remains unchanged throughout the cycle.
3. Since no chemical reaction is taken into account, the working substance is considered as air alone.
4. The specific heats of air are considered as constant, which gives also $c_p/c_v = k = \text{const.}$
5. No heat exchange takes place between the working substance and the engine walls during compression and expansion, these lines being adiabatic curves.

The cycles performed with these simplifying assumptions are called *air-standard cycles*, and the efficiencies computed for them are called *air-standard efficiencies*. In general the efficiency of a theoretical cycle with adiabatic compression and expansion lines is called *thermodynamic efficiency*, and the increase of specific heats with temperature results in a decrease of their ratio k. Taking into consideration this decrease of k, as well as the chemical composition of the gases and the reactions during combustion, gives considerably lower values for the efficiencies of various cycles than the air standard efficiencies. However, the latter are useful because they afford simple means of finding the influence of such factors as the ratio of compression, the amount of fuel admission, the increase of pressure during combustion, and the exponent k of the compression and expansion curves.

5-3. Combination Cycle.—The explosion and the combustion cycles can be considered special cases of the more general combination cycle; therefore the latter will be analyzed first.

Efficiency.—Designating the heat added during the first combustion as $Q_{2\text{-}3}$, Fig. 5-3, and during the second combustion as $Q_{3\text{-}4}$, and the heat rejected at the end of the cycle as $Q_{5\text{-}1}$, the efficiency η can be written as

$$\eta = \frac{Q_{2\text{-}3} + Q_{3\text{-}4} - Q_{5\text{-}1}}{Q_{2\text{-}3} + Q_{3\text{-}4}} = 1 - \frac{Q_{5\text{-}1}}{Q_{2\text{-}3} + Q_{3\text{-}4}} \tag{5-2}$$

where

$$Q_{2\text{-}3} = Wc_v(T_3 - T_2) \tag{5-3}$$

$$Q_{3\text{-}4} = Wc_p(T_4 - T_3) \tag{5-4}$$

$$Q_{5\text{-}1} = Wc_v(T_5 - T_1) \tag{5-5}$$

Expressing all temperatures in terms of temperature T_1 at the beginning of compression gives

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} \tag{5-6}$$

$$T_3 = T_2\,\frac{p_3}{p_2} = T_1\left(\frac{V_1}{V_2}\right)^{k-1}\left(\frac{p_3}{p_2}\right) \tag{5-7}$$

$$T_4 = T_3\,\frac{V_4}{V_3} = T_1\left(\frac{V_1}{V_2}\right)^{k-1}\left(\frac{p_3}{p_2}\right)\left(\frac{V_4}{V_2}\right) \tag{5-8}$$

$$T_5 = T_4\left(\frac{V_4}{V_1}\right)^{k-1} = T_1\left(\frac{V_1}{V_2}\right)^{k-1}\left(\frac{p_3}{p_2}\right)\left(\frac{V_4}{V_2}\right)\left(\frac{V_4}{V_1}\right)^{k-1} = T_1\left(\frac{V_4}{V_2}\right)^{k}\left(\frac{p_3}{p_2}\right) \tag{5-9}$$

To simplify the final expression the following designations are used: For V_1/V_2 the compression ratio r, expression (5-1); for the increase of pressure during the first combustion

$$p_3/p_2 = b \tag{5-10}$$

and for the increase of volume during the second combustion, which will be called the relative heat *admission*,

$$V_4/V_3 = V_4/V_2 = a \tag{5-11}$$

Using expressions (5-3) to (5-11), expression (5-2) for the efficiency becomes after all cancellations

$$\eta = 1 - \frac{a^k b - 1}{r^{k-1}[b - 1 + kb(a - 1)]} \tag{5-12}$$

This expression contains three variable factors, r, a, and b, and shows first of all that the efficiency increases with the increase of the ratio of compression r.

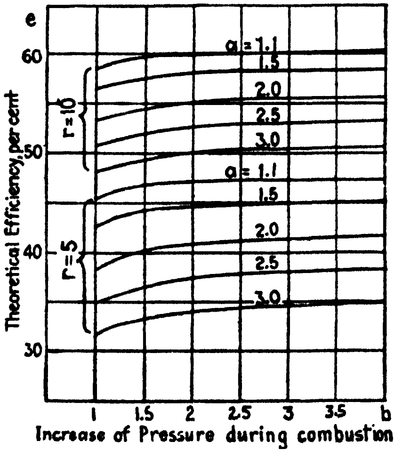

FIG. 5-4.—Effect of r, a, and b on the efficiency of a combination cycle.

The influence of the increase of pressure b can be seen if both the numerator and denominator are divided by b, which gives

$$\eta = 1 - \frac{a^k - 1/b}{r^{k-1}[k(a - 1) + 1 - 1/b]} \tag{5-13}$$

Expression (5-13) shows that with an increase of b the value of η increases. The opposite is true about a—the greater the second heat admission, the lower the efficiency, *i.e.*, the efficiency decreases with the increase of the load on the engine.

To give a better understanding of the efficiencies at different values of a, b, and r, a series of efficiencies were computed using expression (5-12) for a number of values of a and b, approximately covering the ranges as encountered in internal-combustion engines, and for two representative values of r. The results are presented in Fig. 5-4.

As should be expected from expression (5-12), the effect of the compression ratio r is the strongest one, and next comes the adverse influence of the relative admission a. The effect of increasing the pressure during the first part of combustion is comparatively small, especially when the value of b is above 2.

Mean indicated pressure can be expressed, using Fig. 5-3, as

$$p_i = \frac{\left[p_3(V_4 - V_3) + \dfrac{(p_4V_4 - p_5V_1)}{(k-1)} - \dfrac{(p_2V_2 - p_1V_1)}{(k-1)}\right]}{(V_1 - V_2)} \tag{5-14}$$

Expressing p_2, p_3, and p_5 in terms of p_1 by means of expression (2-48) with $n = k$, dividing both the numerator and denominator by V_2, and using expressions (5-1), (5-10), and (5-11), expression (5-14) becomes

$$p_i = \frac{p_1r[b(akr^{k-1} - kr^{k-1} + r^{k-1} - a^k) - r^{k-1} + 1]}{(r-1)(k-1)} \tag{5-15}$$

Expression (5-15) shows that the mean indicated pressure increases with the increase of all three factors, compression ratio r, pressure rise b, and relative admission a.

5-4. Explosion Cycle.—The expression for the efficiency of this cycle can be deduced as a special case of the expression (5-12) for the combination cycle. The absence of the constant-pressure combustion means that, in Fig. 5-3, $V_4 = V_3 = V_2$ and $a = 1$. In this case expression (5-12) becomes

$$\eta = 1 - r^{1-k} \tag{5-16}$$

It is significant that the increase of pressure does not affect the efficiency, *i.e.*, the efficiency of this theoretical cycle is independent of the load and depends exclusively upon the ratio of compression.

In Fig. 5-7 the efficiency curve a of the explosion cycle is shown solid for the range of r actually used in these engines and dotted for higher r. A comparison of curve a and curves in Fig. 5-4 shows that for a given compression r the constant-volume cycle has an efficiency higher than the combination cycle.

Expression (5-15) for the mean indicated pressure with $a = 1$ becomes

$$p_i = p_1 r(b - 1)(r^{k-1} - 1)/(r - 1)(k - 1) \qquad (5\text{-}17)$$

Example 5-1.—Find the relation between the mean indicated pressure and compression ratio for the air-standard cycle, for $r = 4$ to $r = 20$.

In applying expression (5-17), a certain value for b must be used. The latter can be determined from the following considerations: The air charge at the beginning of compression is taken at standard conditions, $p_1 = 14.7$ psi, $t_1 = 60$ F. For the sake of simplicity of calculations the charge may be taken as 1 lb air; the heat ${}_2q_3$ added during the combustion may be taken as equal to that developed by octane if burned with theoretical air in 1 lb air. From Table 4-2

$${}_2q_3 = 19{,}250/15.19 = 1267 \text{ Btu/lb air}$$

The necessary data for $r = 4$ are computed as follows, Fig. 5-1:

$$p_2 = p_1 \times (V_1/V_2)^k = p_1 r^k = 14.7 \times 4^{1.4} = 103.3 \text{ psia}$$
$$T_2 = T_1(V_1/V_2)^{k-1} = 520 \times 4^{0.4} = 905 \text{ R}$$

Assuming a constant average specific heat $c_v = 0.175$ Btu/lb-°F, the weight of the cylinder content is found as

$$W = W_{cA}r/(r - 1) = 1 \times 4/(4 - 1) = 1.333 \text{ lb}$$

and the temperature rise $T_3 - T_2$, by equation (2-24),

$$T_3 - T_2 = 1267/(1.333 \times 0.175) = 5436 \text{ R}$$

Thus $T_3 = 905 + 5436 = 6341$ R; therefore the pressure increase

$$b = p_3/p_2 = T_3/T_2 = 6341/905 = 7.00$$

With these data, the mean indicated pressure, by equation (5-17),

$$p_1 = 14.7 \times 4(7.00 - 1)(4^{0.4} - 1)/(4 - 1)(1.4 - 1) = 218.0 \text{ psi}$$

Repeating these calculations for $r = 8$, 12, 16, and 20, and plotting them, the full-line curve p_i, Fig. 5-5, was obtained.

Figure 5-5 shows a gradual increase of the mean indicated pressure with an increase of the compression ratio. However, when the ratios of p_3/p_i are plotted, the figure shows a rather fast, undesirable increase of the maximum pressure p_3.

In Example 5-1 the heat added was considered constant. If the heat added is taken proportional to the total cylinder content, which, if V is the piston displacement, is equal to $V \times r/(r - 1)$, then the temperatures T_3 will become higher, giving greater b and p_i. The dashed curve in Fig. 5-5 shows this relationship. In terms of an actual internal-combustion engine such a condition will mean that the fresh charge of each cycle is equal not to the volume sucked in by the piston, but to the total cylinder content,

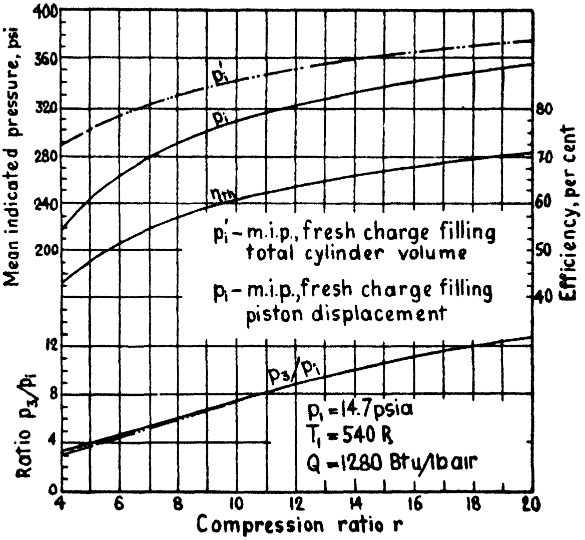

FIG. 5-5.—Effect of compression ratio on the efficiency, mean indicated pressure, and ratio of maximum pressure to indicated pressure in an air-standard explosion cycle.

i.e., to piston displacement plus compression space. Such a condition is obtained by charging the cylinder by outside means, called *supercharging*.

It is interesting to note that, with supercharging, while the absolute value of p_3 also goes up, with compression ratios up to $r = 10$, the ratio p_3/p_i is slightly smaller than without supercharging.

Figure 5-5 also contains a curve of thermal efficiency, computed by expression (5-16).

5-6. Combustion Cycle.—The expression for the efficiency of this cycle can be derived from expression (5-12) by omitting the constant volume combustion. This gives $p_3 = p_2$ and $b = 1$, and expression (5-12) becomes

$$\eta = 1 - \frac{a^k - 1}{r^{k-1}k(a - 1)} \tag{5-18}$$

where, with the designations of Fig. 5-2, $a = V_3/V_2$.

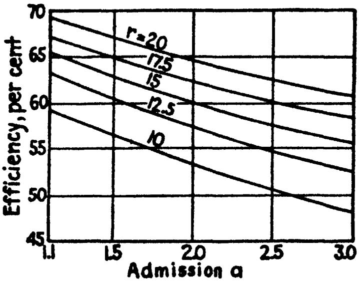

FIG. 5-6.—Effect of a and r on efficiency.

Expression (5-18) shows that the efficiency increases with an increase of r and decreases with an increase of admission a along the same lines as in the combination cycle.

A number of efficiencies figured for different values of r and a are presented in Fig. 5-6. These curves show the pronounced effect of the admission a, or the effect of the engine load upon the efficiency of an engine with a fixed r.

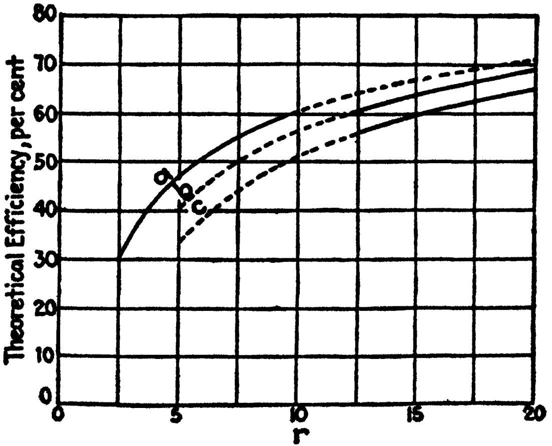

FIG. 5-7.—Efficiencies of explosion and combustion cycles.

To compare the explosion and combustion cycles, curves of efficiency for the latter were drawn in Fig. 5-7: Curve b for a constant admission, $a = 1.5$; and curve c for a constant temperature increase, $\Delta t = 2000$ F at the end of combustion, the admission a varying in this series from 2.8 at $r = 5$ down to 2.2 at $r = 12.5$ and 2.1 at $r = 20$. Below $r = 12$, the cycle is not used in actual engines and the curves are shown dotted. Both curves run below curve a, showing that for equal compression ratios the explosion cycle has a higher efficiency than the combustion cycle. However, the fact that the explosion cycle compresses a combustible mixture instead of an air charge, as does the combustion cycle, limits the former to lower compression ratios, thus lowering its efficiency in actual engines as compared with combustion-cycle engines.

Expression (5-15) for the mean indicated pressure becomes, with $b = 1$,

$$p_i = p_1 r[kr^{k-1}(a - 1) - a^k + 1]/(r - 1)(k - 1) \quad (5\text{-}19)$$

5-6. Temperature-entropy Diagrams.—The various heat changes and the influence of the above mentioned factors, especially of a, b, and r, can be shown very clearly on a T-S diagram. For an air-standard cycle such a diagram, Fig. 5-8, can be constructed, using expressions (2-39) and (2-40) for plotting the lines of $v =$ const. and $p =$ const. and assuming average constant values for c_p and c_v. The zero of entropy may be taken as the point with $t = 60$ F, $p = 14.7$ psia, and a volume v corresponding to 1 lb gas at this t and p. The temperatures at the various points are computed by means of equations (5-6) to (5-9).

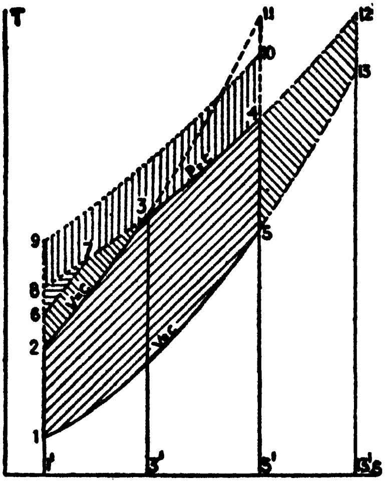

FIG. 5-8.—Entropy diagram.

Area 1-2-3-4-5-1 gives a working diagram of a combination cycle, all points being designated by the same figures as on the indicator card, Fig. 5-3. The heat developed by the first combustion Q_v = 1′-2-3-3′-1′; heat of combustion at constant pressure Q_p = 3′-3-4-5′-3′; heat rejected during exhaust Q_v' = 5′-5-1-1′-5′.

An increase of the compression r raises point 2 to 6 and increases the heat Q_v by area 2-6-7-3-2, which raises the efficiency η of the cycle. An increase up to point 8 eliminates isochoric combustion and gives a cycle with combustion at constant pressure with an increase of Q_v by the area 6-8-7-6 and a higher efficiency. A further increase of r, bringing the end of compression to point 9, increases Q_p and η still more. However, this increase of r is connected with an increase of the maximum pressure which affects the mechanical efficiency adversely.

A continuation of combustion at constant volume from point 3 to 11 changes the combination cycle to an explosion cycle and increases the cycle efficiency.

Finally an increase of admission a from 4 to 12, in order to attain the same temperature as that of the explosion cycle, gives a fast increase of rejected heat Q_v', area 13′-13-5-5′-13′, thus reducing efficiency η in both cases—of a combination and of a constant-pressure cycle.

5-7. Ideal Cycles and Combustion Charts.—The air-standard cycles and their analysis, including the entropy diagram, Fig. 5-8, indicate the influence of various factors upon the performance of internal-combustion engines. However, the simple, hypothetical conditions assumed in this analysis differ so much from conditions in actual engines that the thermal efficiencies thus calculated cannot be used as standards for analyzing the performance of actual engines.

In order to establish usable standards, the concept of an ideal cycle is introduced. An *ideal cycle* may be defined as a cycle of an engine which works under conditions identical with those of an actual engine but has no losses except those due to incomplete combustion, corresponding to conditions of chemical equilibrium, and the loss of energy due to the incomplete expansion of the gases.

Thus an ideal cycle has the following characteristics:

1. The cylinder charge is composed of air, a given amount of fuel, and a certain amount of residual gases from the preceding cycle.
2. All parts of the cycle are adiabatic, and the compression and expansion lines are isentropic.
3. The reactions between the air and fuel follow the laws of chemical equilibrium.
4. The specific heats of the various gases are functions of temperature.

The analysis of different ideal cycles is simplified by the use of compression and combustion charts, Figs. 4-2 to 4-9.

Equivalent mixture is the number of mols M_{eq} of the air-fuel mixture from which the products are formed per 1 lb of air.

If the number of mols of air in the fresh mixture is M_a and that of the fuel is M_f, then, since the molecular weight of air is 28.95, the equivalent mixture

$$M_{eq} = \frac{M_a + M_f}{M_a \times 28.95} \tag{5-20}$$

The values of M_{eq} are indicated on the combustion charts.

Residual Gases.—In an internal-combustion engine the charge consists of the fresh air-fuel mixture and of residual gases left in the compression space from the preceding cycle. The number of mols M_t of the total charge is equal to the sum of mols of the fresh air-fuel mixture M_m and of the residual gases M_r,

$$M_t = M_m + M_r \tag{5-21}$$

With the designation

$$m = M_r/M_m \tag{5-22}$$

the ratio of the mols of the fresh mixture to the mols of the total charge, with the designations (5-21) and (5-22), becomes

$$M_m/M_t = 1/(1 + m) \tag{5-23}$$

The values of chemical energy C used for computing total internal energy U_c and entropy H_c in the charts, Figs. 4-5 to 4-7, are for a pure air-fuel mixture and must be multiplied by the ratio (5-23) or by $(1 - f)$ to take into account the presence of the residual gases, f lb in 1 lb air and residual gases.

The value of m can be determined from the following considerations: The conditions of the residual gases at the end of the exhaust stroke, point 0, Fig. 5-9, are evidently the same as at point 5, expansion extended until $p_5 = p_1 = p_0$; $T_5 = T_0$.

For these two points the characteristic equations are

$$p_5V_5 = M_5RT_5 \tag{5-24}$$

and

$$p_0V_0 = M_0RT_0 \tag{5-25}$$

Combining equation (5-24) and (5-25) and canceling out $p_5 = p_0$ and $T_5 = T_0$ give

$$\frac{M_5}{M_0} = \frac{M_p}{M_r} = \frac{V_5}{V_0} = \frac{V_4}{V_0} \times \frac{V_5}{V_4} = r\frac{V_5}{V_4} \tag{5-26}$$

On the other hand, the mols of products M_p can be presented as

$$M_p = M_m n_2/n_1 + M_r \tag{5-27}$$

where (n_2/n_1) is the ratio of mols of products formed to mols of the air-fuel mixture before combustion. Expressing M_r and M_m in terms of M_p from equations (5-26) and (5-27) and substituting them in equation (5-22) gives

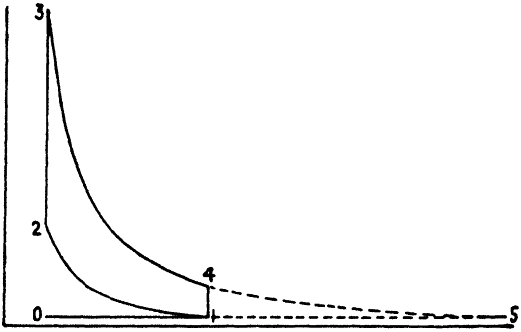

FIG. 5-9.—Complete expansion in an explosion cycle.

$$m = \frac{n_2/n_1}{r(V_5/V_4) - 1} \tag{5-28}$$

Example 5-2.—A mixture of octane vapor with a theoretical amount of air is compressed in an ideal engine which has a compression ratio of 6:1. Conditions at the beginning of the compression stroke are 14.7 psi and 180 F. Determine the conditions at the end of the compression and the work of compression per pound of air.

From Fig. 4-2 the intersection of the curve $p_1 = 14.7$ psi with the horizontal line $T_1 = 180 + 460 = 640$ R gives $V_1 = 16.5$ cu ft and the internal energy $U_1 = 25$ Btu/lb.

With $r = 6.0$, $V_2 = 16.5/6 = 2.75$ cu ft, and an adiabatic (isentropic) line to this volume gives $T_2 = 1185$ R or 725 F, $p_2 = 160$ psi, and $U_2 = 138$ Btu/lb.

The work of compression

$$W_2 = U_2 - U_1 = 138 - 25 = 113 \text{ Btu}$$

Example 5-3.—Determine the ideal efficiency of an explosion-cycle engine, using the data given in Example 5-2.

Assume first a value of m, say $m = 0.044$, and later check it.

From the legend of Fig. 4-2 the chemical energy of the pure mixture is 1280 Btu; and with $m = 0.044$ the chemical energy of the charge is, by equation (5-23),

$$C = 1280/(1 + 0.044) = 1226 \text{ Btu}$$

From Fig. 4-2 or Example 5-2, $U_2 = 138$ Btu and

$$C + U_2 = 1226 + 138 = 1364 \text{ Btu}$$

From Fig. 4-5, when $V = 2.75$ cu ft and $C + U_2 = 1364$ Btu, $T_3 = 5010$ R, $p_3 = 740$ psi, and $S = 0.532$ Btu/R. Adiabatic expansion to $V_4 = 16.5$ cu ft gives $p_4 = 90$ psi, $T_4 = 3590$ R, and $U_4 = 787$ Btu.

Thus the work of expansion is 1364 − 787 = 577 Btu.

From Table 4-3 the heat value of vaporized octane is

$$2{,}368{,}089 - 17{,}730 = 2{,}350{,}359 \text{ Btu/mol}$$

From the reaction, equation (4-24), 1 mol C_8H_{18} requires, with $e = 0$, 59.5 mol air. Therefore the heat value of 1 mol mixture is

$$Q_m = 2{,}350{,}359/(1 + 59.5) = 38{,}870 \text{ Btu}$$

Since the equivalent mixture, as indicated in the legend of Fig. 4-2, is 0.0353 mol, taking into account the presence of residual gases by equation (5-23), the heat value of the charge

$$Q_{ch} = 38.870 \times 0.0353/(1 + 0.044) = 1311 \text{ Btu}$$

The ideal efficiency is the ratio of net work to heat available, or

$$\eta_{id} = \frac{577 - 113}{1311} = 0.354$$

Checking the ratio m of residual gases, isentropic expansion with $S = 0.532$ to $p = 14.7$ gives $V_5 = 68.24$ cu ft.

For the combustion of a theoretically correct mixture of octane and air

$$n_2/n_1 = 64/60.5 = 1.058$$

Therefore, from equation (5-28),

$$m_1 = \frac{1.058}{6 \times 68.24/16.5 - 1} = 0.0444$$

which is within 1 per cent of the value assumed; therefore no recalculations are necessary.

5-8. Ideal Combustion Cycles. *Constant-pressure Combustion.* In an engine operating on this cycle, air and a small amount of residual gases are compressed; the fuel is injected into the highly compressed air and is ignited by its high temperature. In the ideal cycle the fuel is assumed to burn at constant pressure, Fig. 5-2.

The compression process is solved in the same manner as that of an explosion cycle. The internal energy U_2 of the compressed gases at point 2, Fig. 5-2, can be presented as $(M_aU_a + M_rU_r)_2$, where subscripts a and r refer to air and residual gases. Actually the internal energy U_2 may be taken with a sufficient accuracy from Fig. 4-8.

The internal and chemical energies of the liquid fuel injected are M_fU_f and M_fC_f at fuel-supply conditions. The work of injecting the fuel is p_fV_f, where p_f is the injection pressure, in pounds per square foot, and V_f is the volume of the fuel, in cubic feet.

After the constant-pressure combustion, the products will have an internal energy M_pU_p and a chemical energy M_eC_e, where subscripts p and e refer to products of combustion and combustible constituents in the products of reaction at chemical equilibrium. During the combustion the gases expand and do work on the engine piston equal to $p_2(V_3 - V_2)$, Fig. 5-2. The energy equation for the combustion process can be written as

$$
\begin{aligned}
(M_aU_a + M_rU_r)_2 &+ M_f(C_f + U_f) + ApV_f \\
&= (M_pU_p + M_eC_e)_3 + Ap_2(V_3 - V_2) \qquad (5\text{-}29)
\end{aligned}
$$

The left side of equation (5-24) and p_2 are determined from the given conditions. An assumption of V_3 fixes the second term and with it T_3, which permits the evaluation of the term $(M_pU_p + M_eC_e)_3$. If the right side of equation (5-29) does not equal the left one, a new value for V_3 must be assumed and the calculations repeated.

If a combustion chart is available with the corresponding air-fuel ratio, the conditions at the end of the combustion process, V_3 and T_3, are determined by the pressure and energy term.

Example 5-4.—Determine the conditions at the end of a constant-pressure combustion process in an ideal compression-ignition oil engine, using the following data: The pressure and temperature of the charge at the end of the compression are 515 psia and 1440 F, respectively; the ratio of air to residual gases $M_a/M_r = 75$; the fuel is $C_{16}H_{30}$, has a temperature of 60 F, and is injected at 1700 psia pressure; there is an excess air of 50 per cent.

The reaction equation for this example is

$$
\begin{aligned}
C_{16}H_{30} &+ 35.25O_2 + 132.54N_2 \\
&= 16CO_2 + 15H_2O + 11.75O_2 + 132.54N_2 \qquad (5\text{-}30)
\end{aligned}
$$

or

$$1 \text{ mol fuel} + 167.79 \text{ mols air} = 175.29 \text{ mols products}$$

The equivalent mixture for this reaction is, by equation (5-20),

$$M_{eq} = 168.79/(167.79 \times 28.95) = 0.03476 \text{ mol}$$

By equation (5-22)

$$m = 1/75 = 0.0133$$

and by equation (5-23),

$$M_m = 0.03476/1.0133 = 0.03430 \text{ mol mixture}$$

Therefore

$$M_f = 0.03430/168.79 = 0.000203 \text{ mol fuel}$$
$$M_a = 0.03430 - 0.00020 = 0.03410 \text{ mol air}$$
$$M_r = 0.03430 \times 167.79/(168.79 \times 75) = 0.00046 \text{ mol residual gases.}$$

Interpolating from Table 2-2 and computing U_2 for the composition of the products gives

$$(M_aU_a)_{1440} = 0.0341 \times 4812 = 164.2 \text{ Btu}$$
$$(M_rU_r)_{1440} = 0.00046 \times 5200 = \underline{2.4}$$
$$\text{Total energy above 520 R} = 166.6 \text{ Btu}$$

The internal energy U_f of the liquid fuel at 60 F is negative, being numerically equal to the latent heat of evaporation, 30,000 Btu/mol, Table 4-3; and the second term in equation (5-29) becomes

$$M_f(C_f + U_f) = 0.000203(4{,}075{,}000 - 30{,}000) = 821.1 \text{ Btu}$$

The specific gravity of the liquid fuel may be taken as 0.87, and at 60 F the volume of 1 liquid mol is

$$v_f = 222/(62.4 \times 0.87) = 4.17 \text{ cu ft/mol}$$

The work of injecting the fuel is

$$ApV_f = 1700 \times 144 \times 4.17 \times 0.000203/778 = 0.3 \text{ Btu}$$

Thus the left side of equation (5-29) is equal to

$$166.6 + 821.1 + 0.3 = 988.0 \text{ Btu}$$

This is also the value for the right side of equation (5-29).

$$V_2 = \frac{M_2RT_2}{P_2} = \frac{(0.03410 + 0.00046) \times 1545 \times 1440}{515 \times 144} = 1.036 \text{ cu ft}$$

The volume V_3 at the end of combustion is found by a cut-and-try method

Assume V_3, cu ft	2.0	2.5	3.0
$Ap_2(V_3 - V_2)$, Btu	92	140	187
$(M_pU_p + M_eC_e)_3$, by substraction, equation (5-29), Btu	897	849	802
T_3 at 515 psia, Fig. 4-9, °R	4190	4040	3870
V_3 at 515 psia, Fig. 4-9, cu ft	3.125	3.0	2.88

A graphical solution gives at the intersection $T_3 = 3905$ R and

$$V_3 = 2.90 \text{ cu ft,}$$

which in turn gives $Ap_2(V_3 - V_2) = 178$ Btu. An approximate check gives

$$V_3 = \frac{M_3RT_3}{p_3} = \frac{0.03476 \times (175.29/168.79) \times 1545 \times 3905}{515 \times 144} = 2.94 \text{ cu ft}$$

A more accurate check can be made as follows:

$$\begin{aligned} V_3 &= [(M_a + M_f)(M_p/M_m) + M_r]RT_3/p_3 \\ &= [0.0343(M_p/168.79) + 0.000455]1545 \times 3905/(515 \times 144) = 2.90 \end{aligned}$$

Solving for M_p gives $M_p = 173.0$ mols at T_3 and p_3, which is a satisfactory check. The analysis of the other phases of the cycle does not differ from that of the explosion cycle.

Combination Cycle.—The analysis of this ideal cycle, Fig. 5-3, is conducted along the same lines as that of the constant-pressure combustion, but with a modified equation,

$$\begin{aligned} (M_aU_a + M_rU_r)_2 + M_f(C_f + U_f) + ApV_f \\ = (M_pU_p + M_eC_e)_4 + Ap_3(V_4 - V_3) \quad (5\text{-}31) \end{aligned}$$

The following example will illustrate the procedure, particularly when the excess air present at the end of combustion does not correspond to any of the available combustion charts.

Example 5-5.—Determine the conditions at the end of combustion in an ideal compression-ignition oil engine, using the following data: The pressure and temperature of the charge at the end of the compression are 515 psia and 1440 R, respectively; the ratio of air to residual gases $M_a/M_r = 75$; the fuel is $C_{16}N_{30}$, has a temperature of 60 F, and is injected at 1800 psia pressure; the pressure during the second part of combustion is 900 psi; the excess air is 40 per cent.

The combustion-reaction equation gives $n_1 = 157.6$ mols, $n_2 = 164.1$ mols and by equation (5-20), $M_{eq} = 0.03476$ mol. By equation (5-22), $m = 0.0133$, and by equation (5-23), $M_m = 0.03430$ mol mixture; $M_f = 0.000218$ mol fuel, $M_a = 0.03408$ mol air; $M_r = 0.000454$ mol residual gases.

By means of Table 2-2 the internal energy is computed as

$$\begin{aligned} (M_aU_a)_{1440} &= 0.03408 \times 9{,}812 = 164.0 \text{ Btu} \\ (M_rU_r)_{1440} &= 0.000454 \times 5{,}200 = \underline{2.4} \\ \text{Total energy above 520 R} &= 166.4 \text{ Btu} \end{aligned}$$

Similar to Example 5-4, the second term in equation (5-31)

$$M_f(C_f + U_f) = 0.000218(4{,}075{,}000 - 30{,}000) = 882 \text{ Btu}$$

The work of injecting the fuel, similar to Example 5-4, is

$$ApV_f = 1800 \times 144 \times 4.15 \times 0.000218/778 = 0.3 \text{ Btu}$$

Thus the left side of equation (5-31) becomes

$$166.4 + 882 + 0.3 = 1048.7 \text{ Btu}$$

Now $V_2 = \frac{M_2RT_2}{p_2} = \frac{(0.03410 + 0.00045) \times 1545 \times 1440}{515 \times 144} = 1.036$ cu ft

and by Fig. 5-3, $V_3 = V_2$.

The volume V_4 at the end of combustion is found by a cut-and-try method, using combustion charts Fig. 4-9 and, until another chart for $C_{16}H_{30}$ is available, Fig. 4-7.

Assume V_4, cu ft	1.6	1.8	2.0
$Ap_3(V_4 - V_3)$, Btu	93.9	127.3	160.5
$(M_pU_p + M_eC_e)_4$, by subtraction, equation (5-29), Btu	954.8	921.4	888.2
T_4, °R, Fig. 4-9	4390	4280	4160
V_4, cu ft, Fig. 4-9	1.87	1.81	1.78
T_4, °R, Fig. 4-7	4160	4060	3950
V_4, cu ft, Fig. 4-7	1.81	1.78	1.74
T_4, °R, interpolated for $e = 40$ per cent	4333	4225	4107
V_4, cu ft, interpolated for $e = 40$ per cent	1.85	1.80	1.77

A graphical solution gives $T_4 = 4225$ R and $V_4 = 1.80$ cu ft, which in turn gives $Ap_3(V_4 - V_3) = 127$ Btu. A check gives

$$V_4 = \frac{M_4RT_4}{p_3} = \frac{0.03476(164.1/157.6) \times 1545 \times 4225}{900 \times 144} = 1.82 \text{ cu ft},$$

which is close enough.

5-9. Ideal Efficiencies.—For practical applications, ideal efficiencies instead of being computed by means of combustion charts can be taken directly from Figs. 5-10 and 5-11.[1] The data for these curves were computed with the following assumptions:

1. The pressure of the charge at the beginning of the compression is atmospheric, 14.7 psi.
2. The temperature of the fresh mixture was assumed as 100 F, and the temperature of the charge at the beginning of the compression was found, taking into account the temperature of the residual gases.
3. In the explosion cycles the liquid fuel is completely vaporized before it enters the engine cylinder, and combustion occurs at a constant volume.
4. In the combustion cycles, mechanical injection is used, the atomized fuel enters the cylinder at the suction temperature, and combustion occurs at constant pressure.

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[1] LICHTY, L. C., "Internal Combustion Engines," 5th ed., pp. 68*ff*., McGraw-Hill Book Company, Inc., New York, 1939. GOODENOUGH, G. A., and BAKER, J. B., *Univ. Ill. Eng. Exp. Sta., Bull.* 160, 1927.

The efficiencies of the explosion cycles were computed, using as fuel C_8H_{18}.

The efficiencies of the combustion cycles were computed, using as fuel $C_{12}H_{26}$. While the combustion charts use $C_{16}H_{30}$ as engine-fuel oil, the difference is very small. Computations with other hydrocarbon fuels showed that the fuel-composition influence on the efficiency is very small and therefore negligible.

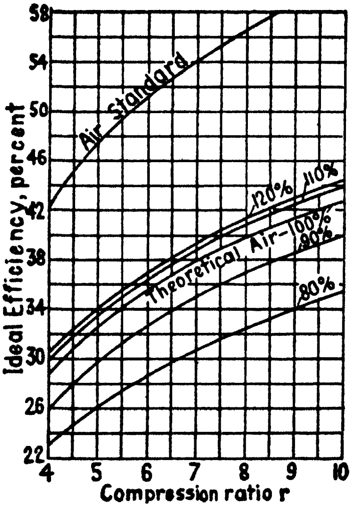

Fig. 5-10.—Explosion cycle.

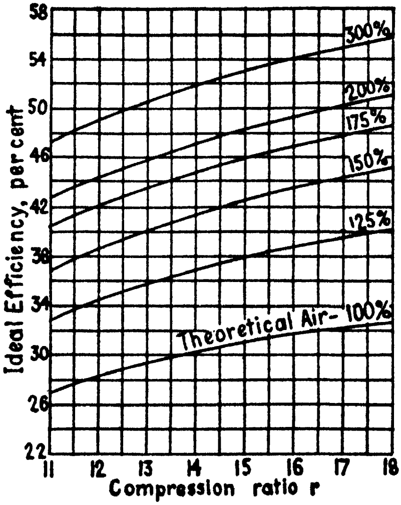

Fig. 5-11.—Combustion cycle.

Figures 5-10 and 5-11 show, in addition to the influence of the compression ratio, that, other conditions being equal:

1. The efficiency increases with the increase of air present.
2. The ideal efficiencies are so much lower than air-standard efficiencies that the latter cannot be used as a basis for comparison of actual engines.

Combination Cycles.—Owing to the presence of two additional variables, the pressure rise b and the admission a, the relations between η_{id}, r, e, a, and b cannot be presented by a single series of curves. The efficiency should be computed for each case, using table data or charts, Figs. 4-8 and 4-9. The approximate value of η_{id} may be estimated from Fig. 5-12, in which the efficiencies are given as functions of the amount of theoretical air for three compression ratios, $r = 14$, 16, and 18, and for three different values of pressure ratios, $p_3/p_1 = 45$, 60, and 75, where

p_3 is the maximum pressure of the cycle and p_1 is the pressure at the beginning of the compression.[1]

For different r and p_3/p_1, the value of η_{id} can be found by interpolation.

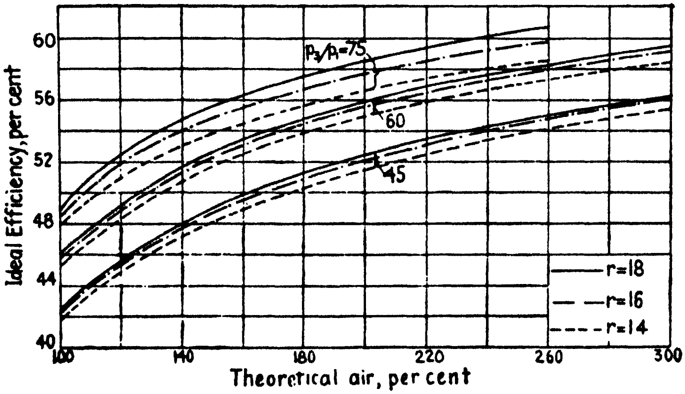

FIG. 5-12.—Effect of theoretical air, compression ratio, and pressure rise on ideal efficiency or a combination cycle.

Example 5-6.—Find the ideal efficiency of a combination cycle having a compression ratio $r = 16$; with a suction pressure of $p_1 = 14.0$ psia and 160 per cent theoretical air the maximum pressure $p_3 = 700$ psi.

The pressure increase is

$$p_3/p_1 = 700/14 = 50$$

Interpolating between the curves $p_3/p_1 = 45$ and $p_3/p_1 = 60$ for $r = 16$, Fig. 5-12, gives for 160 per cent air $\eta_{id} = 50.7$ per cent.

5-10. Problems.—**1.** Find the compression space for a 10-in. × 18-in. engine if the exponent n is 1.37, the initial pressure is 14.0 psia, and if the compression must raise it to 80 psig.

2. Find the relation between the compression ratio, maximum temperatures, maximum pressures, and mean indicated pressures for an air-standard explosion cycle, with $r = 5$, 7, and 9. Plot the results in the shape of curves. Assume standard conditions for beginning of compression; $b = 4$.

3. Find the relation between the compression ratio, maximum temperatures, maximum pressures, and mean indicated pressures for an air-standard combination cycle for $r = 13$, 15, and 17. Assume standard conditions for the beginning of the compression. Referring to Example 5-5, the amount of heat added is based on combustion of $C_{16}H_{30}$, with 50 per cent excess air; assume that the first combustion uses 25 per cent of the fuel; use $c_v = 0.175$. Plot the results in the shape of curves.

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[1] RIXMANN, W., Das Dieselverfahren bei Fahrzeugmotoren, *Z. Ver. deut. Ing.*, vol. 85, No. 5, p. 116, 1941.

4. Work problem 3, assuming 35 per cent excess air and 30 per cent of heat developed during the first combustion.

5. In a combination cycle, the maximum volume is 1 cu ft and the pressure and temperature at the beginning of the compression are 13.7 psia and 140 F, respectively. At the end of the compression the pressure is 100 psia and at the end of the combustion the pressure is 350 psia and the volume 0.3 cu ft. Determine the efficiency on the air-standard basis.

6. The compression space of an 8-in. × 12-in. explosion-cycle engine is 151 cu in. Find the efficiency of the engine: (*a*) on the air-standard basis and (*b*) if the exponent of compression and expansion lines is 1.35.

7. In a theoretical combustion cycle the pressures at the beginning and at the end of the compression are 14.2 and 500 psi, respectively, the exponent of the compression and expansion lines is 1.36, and the piston displacement per stroke is 950 cu in. Find (*a*) the volume of the compression space and (*b*) the efficiency of the cycle, assuming that the volume at the end of combustion is 80 per cent greater than at the beginning.

8. Using combustion charts, find the pressures and temperatures after an adiabatic compression, constant-volume combustion, and complete expansion. The engine has a 5.75:1 compression ratio; the initial conditions are 14.5 psia and 180 F; the fuel mixture is gasoline, C_8H_{18}, with theoretical air.

9. Work problem 8 for a fuel mixture with 10 per cent excess air.

10. Work problem 8 for a fuel mixture with 15 per cent deficiency of air.

11. Using $C_{16}H_{30}$ as fuel, calculate all pressures and temperatures at both dead centers for the following combination compression-ignition engine cycle: Initial conditions are 14.7 psia and 120 F; the compression ratio is 14:1; the increase of the pressure during the first phase of combustion is 50 per cent; the duration of combustion is 7 per cent of the piston stroke after dead center; at the end of the combustion there is 50 per cent of excess air left.

12. Work problem 11 with the following changes: The compression ratio is 16:1, the increase of pressure during the first phase of combustion is 40 per cent, and the duration of the combustion is 6.5 per cent of the piston stroke.

13. Work problem 12 for a 50 per cent increase of pressure during the first phase of combustion.

14. (*a*) using $C_{16}H_{30}$ as fuel, calculate all pressures and temperatures at both dead centers for the following ideal combustion-engine cycle: Initial conditions are 14 psia and 120 F; the compression ratio is 13.5:1; at the end of combustion there is an excess air of 50 per cent; (*b*) determine the relative admission *a* in per cents of the stroke.

15. Work problem 14 with a compression ratio of 14.5:1.

16. Work problem 14 with a compression ratio of 12.5:1.

17. Using data of problem 8, find the ideal efficiency for the cycle.

18. Using data of problem 9, find the ideal efficiency for the cycle.

19. Using data of problem 10, find the ideal efficiency for the cycle.

20. Using data of problem 11, find the ideal efficiency of the cycle.

21. Using data of problem 12, find the ideal efficiency of the cycle.

22. Using data of problem 13, find the ideal efficiency of the cycle.

23. Using data of problem 14, find the ideal efficiency of the cycle.

24. Using data of problem 15, find the ideal efficiency of the cycle.

25. Using data of problem 16, find the ideal efficiency of the cycle.

26. Using Fig. 5-10, find the efficiencies for an ideal engine with a constant-volume combustion and a compression ratio $r = 5.5$ for 80 to 120 per cent theoretical air, and present them in the form of a curve.

27. Work problem 26 for an engine with a compression ratio $r = 6.75$.

28. Find the ideal efficiencies for a combustion-cycle compression-ignition oil engine having a compression ratio of $r = 14.5$ and operating with 150 to 200 per cent theoretical air, and present them in the form of a curve.

29. Work problem 28 for an engine with a compression ratio $r = 13.5$.

30. Using Fig. 5-12, find the efficiencies for an ideal engine with a compression ratio $r = 14$ and pressure rise b varying from 1.12 to 1.40 for different amounts of theoretical air, starting with 120 per cent up to 280 per cent, for every 40 per cent.

31. Work problem 30 for a compression ratio $r = 18$, starting with 100 per cent up to 300 per cent theoretical air, for every 50 per cent.

CHAPTER 6

HANDLING OF GAS FUELS

6-1. Mixing with Air.—The arrangement for mixing fuel gas and air is an important part of a four-stroke gas engine, because complete combustion and low fuel consumption can be obtained only with thorough mixing. The mixing arrangements used in modern engines may be divided into three main groups. In the first group the air and gas are mixed by passing through openings or slots in a separate box and admitted to the cylinder as a ready mixture. In the second group the air and gas are mixed in the throat of a venturi tube by the high velocities created there as a result of the suction action of the engine piston. In the third group the mixing is effected immediately before admission. In most designs the mixing arrangements are connected with the governing devices, and therefore their constructions will be described simultaneously.

Pressure Regulation.—In order to obtain a proper mixing, the fuel-gas pressure should be only high enough to overcome the resistance to the gas flow. For rich, natural gas this pressure should be about 4 in. and not over 7 in. water; for leaner gases it can be slightly higher.

To prevent a fluctuation of the gas pressure due to the intermittent action of the engine piston there should be inserted into the pipe line either a rubber bag or a receiver tank with a capacity of two or three piston displacements. A gasometer consisting of a cylindrical bell floating in water or oil is also often used to take care of the pulsating suction of the engine and to maintain the desired small pressure.

6-2. Governing.—Theoretically the governing of gas engines could be done by two different methods—by changing the weight of the charge without changing its composition and by changing the weight of the fuel gas with practically constant air admission.

The advantage of the first method, called *quantitative governing*, is simplicity—the weight of the charge is changed by throttling both air and fuel gas, or the ready air-fuel mixture; this method of governing can be carried through the whole range of loads, from maximum to idling. The disadvantage is that the lowering of the compression pressure with a decrease of the suction pressure affects the combustion and lowers the thermal efficiency; also the pumping losses increase at smaller loads.

The advantage of the second method, called *qualitative governing*, is better fuel economy. However, this method cannot be used at small loads, because after a certain decrease of fuel admission the mixture becomes so lean that it will not burn properly. Therefore from a certain load down, the air-fuel ratio cannot be increased and the governing must be done by decreasing the total weight of the charge. Thus the governing becomes a combination of both methods, which naturally complicates the construction. This governing is used only in large gas engines where the fuel economy is especially important.

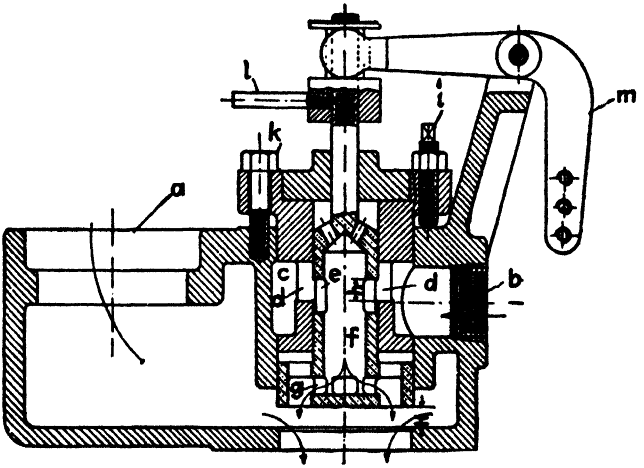

FIG. 6-1.—Gas mixing and regulating valve.

6-3. Gas-air Mixers.—Figure 6-1 shows a mixing box for natural gas. The gas enters at *b* and passes first into the hollow valve stem *f* and from there goes through ports *g* into the air stream. The proportion of gas to air can be adjusted depending upon the gas composition. This adjustment is obtained by slightly turning the valve by the handle *l*, thus changing the size of the effective area of the upper ports which are formed by the edges of slots *d* and *e*.

Governing is done by throttling: The governor connected to the bell crank *m* lowers or raises the valve, changing the heights *H* for air and *h* for gas admission and leaving the ratio of the areas and the mixture unchanged. The drawback of such an arrangement, when used for a two-cylinder engine, is the possibility of an unequal charge being admitted to each cylinder.

In the venturi-type mixer, Fig. 6-2, the air is admitted at *a*, the gas at *b*, led to the throat by the tube *f*. The governor acts upon the butterfly valve *c*. Butterfly valve *d* and the sliding gas valve *e* are hand operated for starting the engine; *g* is gas adjustment for idling. The mixer has a regulator which furnishes the gas at or near atmospheric pressure. This type of mixer used with natural gas gives very good results over the entire range of load and speed variations. As compared with mechanical mixers such as shown in Fig. 6-1, the venturi type is less expensive, more easily adjusted, and does not require as powerful a governor.

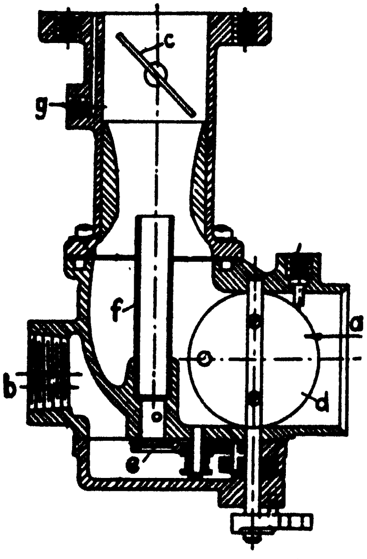

FIG. 6-2.—Venturi gas mixer.

An example of a construction with a mixing arrangement of the third type but with the same governing method by throttling is shown in Fig. 6-3. Air is admitted through a balanced double-disk valve *a* and the gas through a conical-plug valve *b*. Mixing is effected by means of a number of nozzles *c*, *c*. The proper air-fuel ratio is adjusted by slightly lowering or raising the gas valve on its stem. The charge is regulated by simultaneously lowering or raising the air and gas valves. Sliding surfaces, friction, danger of sticking, and reaction upon the governor are eliminated in this design. The use of separate regulating valves for each cylinder and elimination of an admission manifold permit the adjustment of the proper charge for each cylinder.

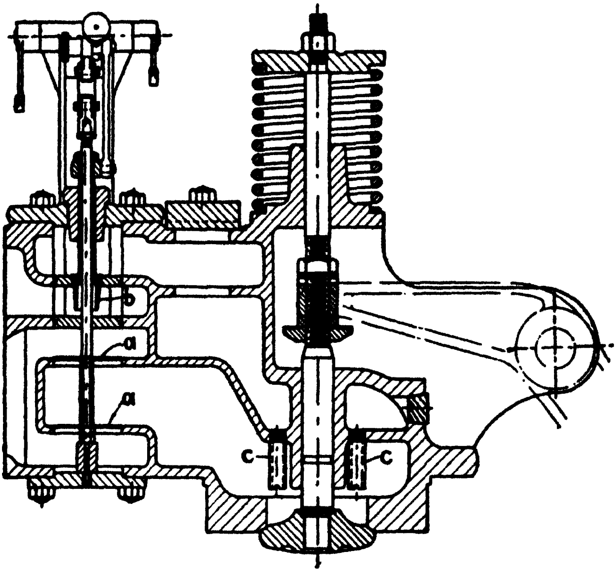

Fig. 6-3.—Gas mixing, regulating, and inlet valves.

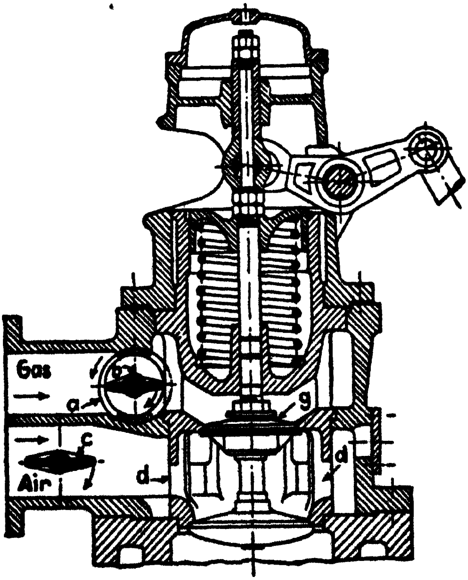

Fig. 6-4.—Inlet valve and mixing arrangement with combination governing.

Figure 6-4 illustrates a mixing arrangement of the third group with a combination governing. From maximum load down to about half load the governing is done by throttling of the fuel gas alone by means of the cylindrical valve *a*; after that both air and fuel are throttled by means of butterfly valves *b* and *c*. The mixing is assisted by the air stream being broken by ribs *d*. However, this system of mixing can be used only with lean gases such as blast-furnace or producer gas.

For richer gases, when the air-fuel ratio becomes larger, as shown in Table 3-1, a finer breaking up of the gas and more intimate mixture of gas in air are necessary.

Two-stroke Engines.—In two-stroke gas engines, in which the scavenge air is admitted through ports uncovered and covered by the piston, the gas is admitted through similar separate slots and the mixing occurs in the cylinder itself. In one modern American engine operated on natural gas, the gas is allowed to enter into the cylinder only after the piston has covered both the scavenge and exhaust ports. Gas is admitted, or as the engine builders say, is *injected*, under a comparatively high pressure through a nozzle. This method avoids loss of gas into the exhaust, and the engines show a considerably smaller fuel consumption per horsepower-hour.

Propane Mixers.—Before use in an engine, propane, which is a liquid under a high pressure, must be evaporated by reducing the pressure and by heating it with a steam coil.[1] The gas has a heat value of 2330 Btu/cu ft and for better mixing in the engine is first diluted with air in order to obtain a mixture of about 1500 Btu/cu ft. This propane-air mixture can be used instead of natural gas in an engine with a conventional air-fuel mixing arrangement.

6-4. Compression-ignition Gas Engines.—These engines are built as convertible engines to use either gas or liquid fuel. The conversion is made with minor changes only. The same piston, cylinder heads, connecting rods, and injection nozzles are used with either fuel; all the equipment used for liquid fuel is retained when operating on gas.

Figure 6-5 gives the schematic arrangement for a two-stroke engine. The cylinder is filled with air obtained from a scavenge pump and compressed to about 475 psig. At the end of the compression stroke, gas is injected through the regular fuel-injection valve and ignited by the heat of compression. A small amount of fuel oil, called *pilot oil*, is injected simultaneously in order to give smoother burning. The gas to be injected is compressed in a three-stage compressor of the same design and dimensions as the compressor used for air injection of liquid fuel. The final gas pressure is 1050 to 1200 psi. The gas is cooled between stages and is discharged into a common line. From this line the gas is admitted through needle valves to the individual cylinders. The lift and duration of the injection are controlled by a governor through so-called *fuel-valve actuators.*

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[1] *Diesel Power and Transportation*, vol. 18, No. 9, pp. 767–769, September, 1940.

Each cylinder is equipped with an individual Bosch fuel pump. When running on liquid fuel, this pump sends the oil to the fuel-injection valve. When the engine is running on gas, the pump *p*, Fig. 6-5, is connected with the hydraulic fuel-valve actuation. The fuel pump *p* then pumps lubricating oil instead of fuel oil; however, the oil in the line between the fuel pump and the valve actuator *a* does not circulate except for a slight amount of leakage; a water jacket *j* cools this line. The hydraulic valve-actuator plunger *e* is connected to the injection-valve lever, which lifts the needle valve *f*.

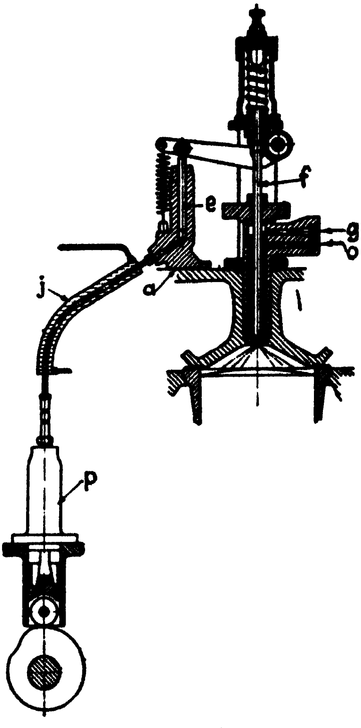

Fig. 6-5.—Schematic arrangement of a compression-ignition engine for running on gas. (*Courtesy of Nordberg Manufacturing Company.*)

Each Bosch pump, in turn, is connected to the governor in the usual manner, so that the amount of oil delivered by the pump, and hence the movement of the plunger, is in proportion to the engine load. The pilot oil is delivered to the injection valve at *o* by an additional small multiple Bosch pump driven from the cam shaft. The amount of pilot oil per cycle remains constant and corresponds to about 5 per cent of the total Btu of the fuel consumed at full load. Gas is admitted at g.

When the engine is running on fuel oil, the gas compressor is either used as an air-injection compressor or, if the engine is operated with airless injection, it is simply disconnected.

Lately four-stroke compression-ignition engines of several makes have been developed which can be changed over from fuel-oil to gas operation or back while the engine is running, even under load.[1] Basically the change-over is effected simply by opening or closing a gas valve and letting the governor regulate the fuel-oil admission. These engines are called *dual-fuel* engines. The fuel combination can be varied, using any proportion of oil and gas, from 5 per cent oil and 95 per cent gas to operation on fuel oil alone. The minimum fuel-oil admission acts as a pilot flame to ensure ignition and smooth burning of the gas, particularly at small loads.

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[1] *Diesel Power and Transportation*, vol. 23, No. 3, pp. 253, 299, and 300, March, 1945.

CHAPTER 7

HANDLING OF VOLATILE LIQUID FUELS

7-1. Principles of Carburetion.—The formation of an inflammable mixture of air and a finely atomized light hydrocarbon is called *carburetion.* The atomization is accomplished by discharging the liquid hydrocarbon through small-hole nozzles or jets into a stream of moving air. The suction stroke of an engine creates a vacuum in the cylinder and produces a gradual pressure decrease from the atmospheric pressure outside the carburetor to the pressures in the carburetor, manifold, and engine cylinder. This pressure drop causes air to flow through the induction system and the fuel to be sprayed from the fuel jets. The pressure drop at the fuel-jet tip is increased by placing it in the throat of a venturi tube, which increases the air velocity and pressure drop at this point. The vacuum at the venturi throat is called the *carburetor depression.*

The stream of fuel leaving the jet is broken up, because of the pressure difference, into a spray consisting of droplets of various sizes. Because of the volatility of the fuel, vaporization from the surfaces of the droplets takes place; this causes the disappearance of the finest droplets and the reduction in size of the others. While it is desirable to have all the droplets vaporized and the vapor uniformly distributed in the air before the mixture enters the engine cylinder, the larger droplets usually reach the cylinder in liquid form and must be vaporized and mixed with air during the compression stroke.

The degree of atomization depends upon the relative velocities of the air and fuel stream, the depression and physical characteristics of the fuel, and its density and surface tension. For a given fuel the degree of atomization is increased with the square of the relative velocity.

Carburetor Requirements.—*Carburetors* are devices used in internal-combustion engines to discharge into the air stream the desired quantity of liquid fuel, to atomize it, and to produce a homogeneous air-fuel mixture. Carburetors are constructed chiefly for gasoline fuel, though some are made for such volatile fuels as kerosene or alcohol.

Automotive engines, operating under variable speed and load conditions, present the most difficult requirements to carburetors. These are:

1. Ease of starting, particularly under low-temperature conditions.
2. Ability to give full power quickly after starting the engine.
3. Equally good, smooth operation at varying loads, absence of racing when idling, and full torque at low speeds.
4. Good acceleration.
5. Sufficient power at high engine speeds.
6. Good fuel economy.

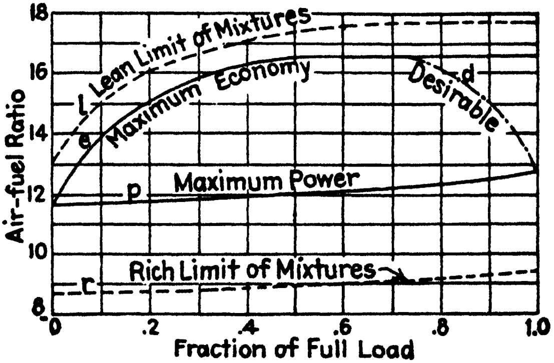

FIG. 7-1.—Ranges of air-fuel mixtures.

Since the operation of a gasoline engine depends primarily on the quality and quantity of the air-fuel mixture delivered to its cylinders, the above requirements can be summed up in one statement: A good carburetor must produce the desired air-fuel ratio at all speeds and loads and do it automatically.

Air-fuel Ratios.—In stationary engines the desired air-fuel ratio is the one that gives the maximum economy. In engines used for transportation, whose speed must be changed within wide ranges, the air-fuel ratios must also change, depending on whether maximum economy or maximum power is desired. At any engine speed the load may vary from no load to full load, depending on the position of the throttle, but no-load and full-load conditions are different at different speeds.

Designating the full load at a certain speed as unity and no load as zero, the intermediate loads will appear as fractions of full load. Using loads as abscissas and air-fuel ratios as ordinates, curve p, Fig. 7-1, of mixtures giving maximum power and curve e of maximum economy, may be plotted.[1] Curve l for limits of lean mixtures and curve r for limits of rich ones for proper firing complete the picture. The low air-fuel ratios, or rich mixtures at small loads, are required because of the dilution of the charge by residual gases. As will be shown later, the dilution increases with a decrease of the suction pressure, which always takes place at small loads.

Maximum power is needed near and at full load; therefore the desirable mixtures should follow the curve e of best economy up to about 0.7 full load and then bend down, curve d, and reach the maximum-power curve p at full load.

7-2. Flow of Fluids.—The fundamental equation for the flow of fluids, both liquid and gaseous, is

$$v = \varphi\sqrt{2gh} \tag{7-1}$$

where v is the velocity of flow, fps,
φ is the velocity coefficient,
g is the acceleration due to gravity = 32.2 fpsps,
h is the head of fluid causing the flow, ft.

The quantity of fluid flowing is given by the expression

$$W = w\alpha A v = wA\mu\sqrt{2gh} \tag{7-2}$$

where W is the weight of fluid flowing, lb/sec,
w is the specific weight of the fluid, lb/cu ft,
A is the cross area of the passageway, sq ft,
α is the coefficient of contraction, about 0.95,
μ is the coefficient of discharge $\mu = \varphi\alpha$.

In carburetor computations the head h is expressed in h_w, inches of water, and

$$h = h_w w_w / 12w \tag{7-3}$$

where for average temperatures the specific weight of water $w_w = 62.4$. Substituting h from equation (7-3) in equation (7-2) gives

$$W = 18.3A\mu\sqrt{h_w w} \tag{7-4}$$

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[1] TICE, P. S., Fuel-supply Requirements, *Automotive Ind.*, vol. 53, p. 53, 1925.

Using this expression for both air and fuel gives

$$W_a = 18.3 A_a \mu_a \sqrt{h_w w_a} \qquad (7\text{-}5)$$

where w_a is in the venturi throat, and

$$W_f = 18.3 A_f \mu_f \sqrt{h_w w_f} \qquad (7\text{-}6)$$

The more accurate equation for the flow of air through a venturi with $c_p/c_v = 1.41$ is

$$W_a = 178.5 A_a \mu_a \sqrt{(p_1/v_1)[(p_0/p_1)^{1.42} - (p_0/p_1)^{1.71}]} \qquad (7\text{-}7)$$

where p_1 and v_1 are the pressure in pounds per square inch and specific volume in cubic feet, respectively, at the entrance, and p_0 is the pressure in the throat.

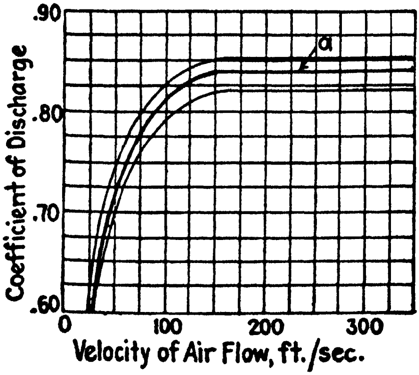

FIG. 7-2.—Venturi discharge curves.

However, expression (7-5) is much simpler, and its accuracy is sufficient if the pressure difference $(p_1 - p_0)$ is small and the coefficient μ_a is properly selected.

If the pressure difference is greater than 50 in. water, equation (7-7) should be used. The maximum discharge W is reached when $p_0 = 0.53 p_1$ and a further decrease of the back pressure p_0 does not increase W.

Coefficients of Discharge.—The coefficients of discharge μ_a for air through pipes and venturi tubes vary from 0.60 to 0.98. The venturi tubes used in carburetors have a coefficient μ_a, which lies between 0.82 and 0.85, depending upon the shape and finish of the tube, but is practically constant from about 150 fps and up, with a fair average of 0.84, curve *a*, Fig. 7-2. For velocities below 150 fps, μ_a decreases very rapidly.

Good practice is not to exceed 150 fps, at 1000 rpm, in the venturi.[1]

The coefficients of discharge μ_f for liquid fuels depend upon many factors:

1. Size of orifice.
2. Ratio of length to diameter.
3. Shape of orifice—circular, annular, chamfered, etc.
4. Entrance to orifice.
5. Viscosity of the fuel which varies with temperature.
6. Head causing the flow or rate of flow.

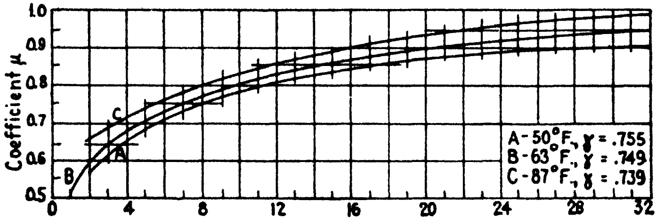

FIG. 7-3.—Coefficient of discharge of gasoline at different heads and temperatures.

In a small orifice the friction surface is considerable compared with the volume. With the increase of orifice diameter, the ratio of surface to volume decreases and the coefficient μ increases.

The ratio of length l to diameter d influences μ for the same reason; the smaller l/d, the greater μ becomes.

An annular orifice presents more friction area than a circular one and consequently gives a lower coefficient μ. Chamfering increases the coefficient considerably.

A well-rounded entrance to the fuel tube increases the coefficient μ.

The influence of viscosity is obvious—the higher the viscosity, the greater the resistance, the smaller the discharge, and the lower the coefficient. For a given fuel, viscosity decreases with increase of temperature and the coefficient μ becomes greater.

The coefficient μ increases with an increase of the head causing the flow.

Coefficients of discharge, for fuel jets with circular orifices, range from 0.6 to 1.0, and even higher, depending upon the different factors and working conditions. Figure 7-3 gives values of μ

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[1] *SAE Journal*, vol. 50, No. 6, p. 252, 1942.

for a typical carburetor orifice and shows the influence of head, causing the flow, and temperature of the fuel changing the specific gravity γ.[1]

7-3. Carburetors. *Simple Carburetor.*—The simplest carburetor, Fig. 7-4, would consist of a fuel jet *j* located in an air passage *a*, a chamber with a float *f* for maintaining the level in the jet, and a throttle valve *t* for controlling the amount of mixture delivered to the engine.

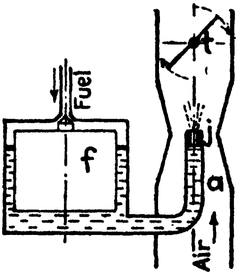

FIG. 7-4.—Simple carburetor.

When the air flows through the venturi, it creates a slight vacuum at the throat where the jet is located; this causes the fuel to rise and to be discharged into the air stream. However, the vacuum must reach about 0.35 in. of water before the fuel will flow from the jet. This vacuum, or *depression*, is necessary to overcome the viscosity of the fuel and also, because the level of the fuel is slightly below the top of the jet, to prevent spilling. To obtain the necessary depression even at low air velocities the air passage near the jet is contracted, forming a venturi, which gives an increase of velocity with a very small resistance.

TABLE 7-1.—CARBURETOR INTAKE

Depression h_w, in. water	Δp, psi	Pressure, psia	Density w_a at 60 F, lb/cu ft	Product, $h_w w_a$
0	0.0	14.70	0.0764	0.0
10	0.36	14.34	0.0745	0.745
20	0.72	13.98	0.0726	1.452
30	1.08	13.62	0.0708	2.124
40	1.44	13.26	0.0688	2.752
50	1.80	12.90	0.0670	3.350
60	2.16	12.54	0.0652	3.912

Designating the air-fuel ratio by r_{af},

$$r_{af} = W_a/W_f \qquad = \frac{h - fG}{f \quad Pr} \qquad (7\text{-}8)$$

and using the above derived expressions (7-5) and (7-6) but introducing in the latter the value of $h_w - h_w'$ instead of h_w, where h_w' as stated is approximately 0.35 in., gives

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[1] Based on data from *NACA Rept.* 49, 1919.

$$r_{af} = (A_a\mu_a/A_f\mu_f)\sqrt{w_a h_w/w_f(h_w - h_w')} \qquad (7\text{-}9)$$

With the increase of the engine load and a gradual opening of the throttle, the quantity of air going into the engine must increase. This increase is the result of an increase of the depression h_w, as may be seen from expression (7-5). On the other hand, with a decrease of the pressure, the air density w_a decreases, Table 7-1, and, according to equation (7-9), r_{af} decreases. Thus the mixture becomes richer with an increase of the load, contrary to the requirements as shown in Fig. 7-1.

Example 7-1.—A $3\frac{1}{2}$-in. × $4\frac{1}{2}$-in. eight-cylinder four-stroke engine running at 2500 rpm has a carburetor with a $1\frac{1}{4}$-in. throat. Determine the depression in the venturi throat at a standard outside pressure if the throttle produces a 75 per cent volumetric efficiency.

The volume of air drawn in cubic feet per second:

$$V_a = \frac{0.7854 \times (3.5)^2 \times 4.5 \times 8 \times 2500 \times 0.75}{2 \times 1728 \times 60} = 3.13 \text{ cu fps}$$

Anticipating a depression of about 10 per cent or 1.4 psi, and taking, according to Table 7-1, $w_a = 0.0688$, the weight is found approximately as

$$W_a = 3.13 \times 0.0688 = 0.2152 \text{ lb/sec}$$

The average air velocity in the throat is $\frac{V_a}{A} = v$

$$3.13/[0.7854 \times (1.25)^2/144] = 3.13/0.00853 = 367 \text{ fps}$$

From Fig. 7-2 for this velocity, $\mu_a = 0.84$, and from expression (7-5)

$$h_w w_a = (W_a/18.3A_a\mu_a)^2 = [0.2152/(18.3 \times 0.00853 \times 0.84)]^2 = 2.70$$

Interpolating from Table 7-1 gives the corresponding $h_w = 39.17$ in. depression and $w_a = 2.701/39.17 = 0.069$, which is very close to the previous assumption.

Example 7-2.—Determine the diameter of a fuel orifice to give a 14:1 air-fuel ratio in the previous example for a simple one-jet carburetor using gasoline with a specific gravity 0.72.

In accordance with Fig. 7-3, assume the coefficient of discharge of the fuel orifice $\mu_f = 0.93$. Then from equation (7-9)

$$A_f = \frac{A_a\mu_a}{r_{af}\mu_f}\sqrt{\frac{w_a h_w}{w_f(h_w - h_w')}} = \frac{0.00853 \times 0.84}{14 \times 0.93}\sqrt{\frac{0.069 \times 39.17}{0.72 \times 62.4(39.17 - 0.35)}}$$

$$= 0.00002163 \text{ sq ft}$$

$$D_f = \sqrt{0.00002163 \times 144/0.7854} = 0.063 \text{ in.}$$

Example 7-3.—Determine the air-fuel ratio if with an increase of the load the depression increases to 50 in. in the previous example.

According to equation (7-9), if h_{w2} is the new depression, then, with w_{a2} equal 0.067 from Table 7-1, with a sufficient accuracy

$$r_{af}' = r_{af}\sqrt{w_{a2}/w_a} = 14\sqrt{0.067/0.069} = 13.79$$

Compensated Carburetors.—To compensate for the tendency of a simple jet to increase the richness of the charge with increase of load, several devices are used:

1. An auxiliary air valve that automatically admits additional air as the mixture flow increases.
2. A compensating jet that allows an increasing flow of air through a fuel passage as the mixture flow increases.

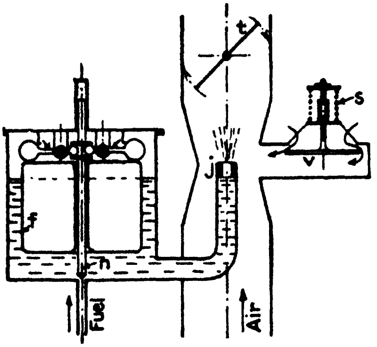

FIG. 7-5.—Carburetor with auxiliary air valve.

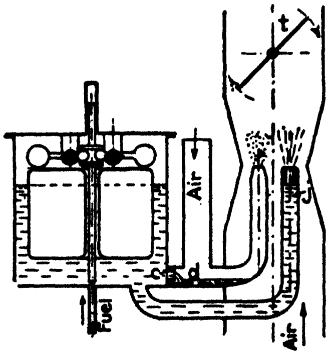

FIG. 7-6.—Carburetor with compensating jet.

3. A tapered metering pin that reaches into the fuel nozzle and is lifted or lowered, thus changing the quantity of the fuel drawn into the air charge.
4. A combination of a variable air passage and a tapered metering pin, controlled by the air valve to give the desired mixture for any load.

Figure 7-5 shows an auxiliary-valve carburetor. The valve spring *s* is operated by the vacuum, which increases with increase of engine load and increases the air admission in proportion to the lift of the valve *v*. Figure 7-5 also shows the action of the float chamber less schematically than Fig. 7-4. The fuel needle *n* is connected to the float *f* by levers, *l*, *l*; when the float rises, the needle is pushed down closing the fuel admission. This arrangement allows a more convenient connection between the carburetor and the fuel-supply tank from below.

Figure 7-6 shows diagrammatically the Zenith carburetor with a compensating jet *c*, which discharges fuel at a constant rate into a well *d*, open to the atmosphere. The main jet *j* is made to give a mixture which is at all times too lean, but becomes richer as the engine speed and load increase. The compensating jet *c* alone would give a mixture that is also too lean and that becomes still weaker, as the engine speed and load increase as more air is drawn in from well *d* and a constant amount of fuel is discharged by gravity. The two jets working together, properly proportioned, compensate one another and give a mixture which may be constant or become leaner or richer with increase of load *T*, depending upon the design.

Idling Carburetor.—The idling- and low-load carburetor discharges the air-fuel mixture at the edge of the throttle plate in its closed position, Fig. 7-7. With the throttle closed, the manifold depression may rise to 200 in. of water, but the depression at the idling jet *i* will be considerably smaller because of the admission of air through the idling adjustment *a* and because of a certain air flow past the edge of the throttle plate *t*. The desirable idling air-fuel ratio is obtained by adjusting the idling screw *a*.

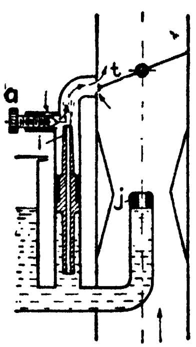

Fig. 7-7.—Carburetor with idling jet.

The manifold depression decreases as the throttle is opened until, at a wide-open throttle and low speed, the depression may drop to about 5 in. of water or less. As the depression decreases, the amount of fuel delivered by the idling jet *i* gradually decreases to practically zero.

Acceleration.—With a partial throttle opening, the pressure in the intake manifold is appreciably reduced and the fuel is vaporized quite readily.[1] At low speeds, when the throttle is wide open to accelerate the engine, the heavier fractions of gasoline will begin to condense and collect on the walls of the intake manifold, forming a fuel film. This results in a leaner air-fuel mixture entering the cylinder. When the wall film is built up, the mixture returns to the desirable air-fuel ratio. With a sudden acceleration of the engine the mixture would become so lean that firing back in the carburetor or failure of the mixture to ignite would result, unless extra fuel is furnished for acceleration.

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[1] Tice, *loc. cit.*

The following devices to provide this extra fuel may be used:

1. A damped air valve.
2. An accelerating well.
3. A displacement pump operated by the throttle.
4. A displacement pump operated by suction.

The air valve, even with very little damping, ordinarily delivers too much accelerating fuel. The accelerating well, Fig. 7-6, has the same defect, although to a smaller degree. The throttle-operated pump usually furnishes less than the average desirable amount. The suction-operated pump furnishes about the desirable amount for partial acceleration but not enough for full-load conditions. A combination of a suction-operated and a throttle-operated pump, which comes into action as the throttle approaches the wide-open position, gives the best results, approaching closely the desirable curve *d*, Fig. 7-1.

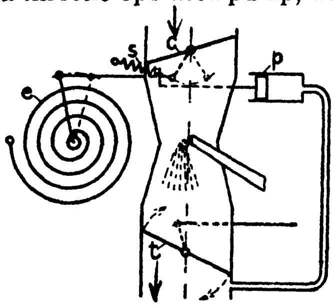

FIG. 7-8.—A downdraft carburetor with automatic choke.

Choking.—Starting in cold weather requires a very rich air-fuel mixture which is induced by choking. A choke is a butterfly valve, similar to the throttle valve *t*, Fig. 7-4, but located before the venturi. Choking increases the depression in the venturi and causes an oversupply of liquid fuel. At low temperatures only the lighter fraction will evaporate, giving a small amount evaporated x, but the low air-fuel ratio r_{af} gives an ignitible air-vapor mixture r_{av}.

Choking should be limited to the minimum necessary to obtain a starting mixture, as the unvaporized heavier fractions of gasoline wash off the lubricating oil from the cylinder walls and, running down into the crankcase, dilute the oil.

In modern engines choking is done automatically by a thermostat *e*, Fig. 7-8, which, aided by the spring *s*, closes the choke *c* when the engine is not running. As soon as the engine starts, the manifold vacuum causes the vacuum piston *p* to open the choke *c*. Exhaust gases heat the thermostat casing and bimetallic spring and allow the choke to assume a wide-open position.

A fast-idling device, usually incorporated in the automatic choking mechanism, keeps the throttle partly open and decreases the warming-up period.

Commercial Carburetors.—A good commercial carburetor, particularly for use with automotive engines, contains in its design all the above described devices: (*a*) A compensating device to obtain automatically the desirable air-fuel ratio under all load and speed conditions; (*b*) an automatic choke for starting; (*c*) an idling carburetor; (*d*) a device for providing additional fuel for acceleration; and (*e*) a device for increasing the fuel delivery at wide-open throttle. Actual carburetor characteristics obtained from engine tests, Fig. 7-9, show how closely the theoretically desirable fuel ratios, Fig. 7-1, are approached in practice, both as a function of load variation, Fig. 7-9*a*, and at different speeds, Fig. 7-9*b*.

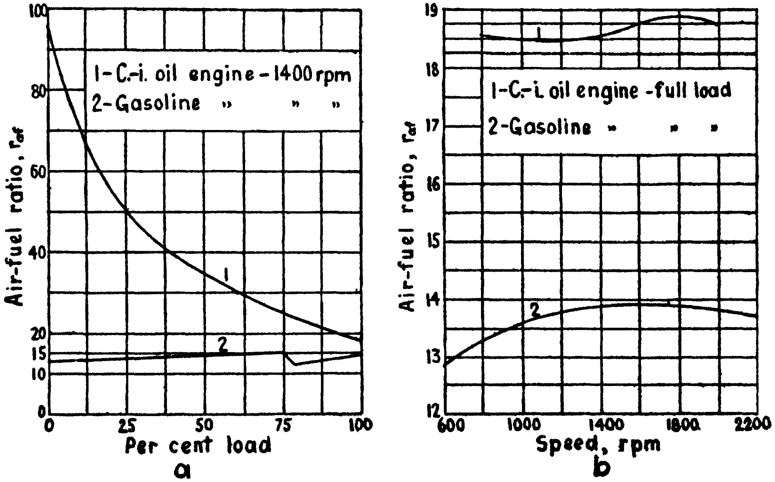

FIG. 7-9.—Air-fuel ratios in a compression-ignition oil engine and in a gasoline engine at different loads and speeds.

Constructions.—The existing carburetors, in respect to the location of the venturi tube, can be divided into three groups: (*a*) vertical, (*b*) horizontal, and (*c*) inverted carburetors. In a vertical or *updraft* carburetor the air-fuel mixture flows upward, as in Figs. 7-4 to 7-6. This arrangement is convenient for a gravity flow of the fuel and gives a low over-all height.

A horizontal carburetor is very accessible and gives a higher charge efficiency because of shorter connections to the intake manifold. On the other hand, it takes more space and is not very suitable for an air filter.

The inverted or *downdraft* carburetor, as in Fig. 7-8, has for its main advantages great accessibility and ease for attaching an air filter or silencer. Another advantage is that all the fuel, including the liquid particles, has a tendency to flow toward the manifold, because of the force of gravity, and from there to the cylinders; there is no danger of the settling out of the heavier fractions of the fuel.

The design and construction of carburetors for automobile engines as well as for other purposes at present form a special, separate industry. Therefore it is not necessary to go into more detailed discussion of carburetor problems here, outside of mentioning that special carburetors are built for kerosene and alcohol.

7-4. Aircraft-engine Carburetors.—These engines operate under varying conditions of atmospheric pressure and temperature, depending upon the altitude of the flight. Special provisions must be made in order to prevent the formation of an unnecessarily rich mixture causing waste of fuel, as the amount of air drawn into the engine will decrease with the altitude.

Effect of Altitude.—At sea level the average or standard atmospheric pressure is 14.7 psia. With an increase of altitude the atmospheric pressure decreases, and as a result an engine receives less air per stroke. The relation between the atmospheric pressure p, psia, and altitude h, ft, above sea level is expressed as[1]

$$h = 122.86T_m \log_{10} (14.7/p) \tag{7-10}$$

where T_m, °R, is the mean temperature between T_s at sea level and T at the altitude h. For a given h, the relation is conveniently presented as

$$p = 14.7/\text{antilog}_{10} (h/122.86T_m) \tag{7-11}$$

The temperature of the air decreases with altitude approximately 3.566 F per 1000 ft,

$$t = t_s - 0.003566h \tag{7-12}$$

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[1] DIEHL, W. S., Standard Atmosphere, Tables and Data, *NACA Rept.* 218, 1925.

The decrease of the pressure decreases the specific weight of the air, and the decrease of temperature increases it. However, the decrease due to altitude is greater than the increase due to temperature drop and, according to equation (7-9), the air-fuel ratio will decrease with the altitude.

Example 7-4.—Determine the air-fuel ratio at 15,000 ft altitude in a carburetor adjusted at sea level for a 15.2:1 ratio; the air temperature at sea level is 68 F, and the barometric pressure is 14.7 psi.
The temperature, according to equation (7-12),

$$t = 68 - 0.003566 \times 15{,}000 = +14.5 \text{ F}$$

and $T_m = (68 + 14.5)/2 + 460 = 501\text{R}$
Using equation (7-11) gives for the pressure

$$p = 14.7/\text{antilog } (15{,}000/122.86 \times 501) = 14.7/1.754 = 8.39 \text{ psia}$$

The air density

$$w_{a2} = \frac{0.0763 \times 8.39 \times 520}{14.7 \times (460 + 14.5)} = 0.0478 \text{ lb/cu ft}$$

According to expression (7-9) and Example 7-3,

$$r_{af}' = r_{af}\sqrt{w_{a2}/w_a} = 15.2\sqrt{0.0478/0.0763} = 12.03$$

or 25 per cent excess of fuel which will be wasted.

Compensated Carburetors.—The effect of altitude must be compensated by admitting either more air or less fuel. To accomplish this one of the following methods can be used:

1. An auxiliary air valve or air port.
2. Changing the position of the jet in the venturi.
3. A variable fuel-metering orifice with a taper pin.
4. A pressure reduction in the float chamber.

Figure 7-10 illustrates the first method. By opening the butterfly valve *b*, additional air is admitted and at the same time the depression in the venturi is reduced, decreasing the quantity of gasoline drawn in.

The suction action is highest in the throat of a venturi; therefore by raising the venturi, Fig. 7-11, the jet relatively moves to points with smaller suction and the flow of fuel is thus decreased.

Both these devices may be operated manually or preferably automatically by means of a bellows of corrugated sheet metal similar to an aneroid barometer.

A variable fuel-orifice carburetor, with throttle segments instead of a butterfly valve, is shown in Fig. 7-12. These segments *s*, *s* are shaped to provide a variable rectangular venturi and are rotated through part of a circle from closed to the wide-open position, shown in dotted lines. The needle valve is closed when the throttle is closed, and idling fuel flows through the idling orifice *i* and then through the main fuel orifices, together with the bled air from *b*, *b*.

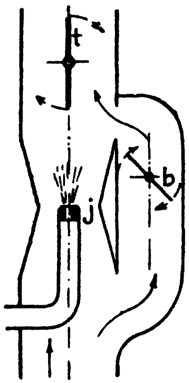

FIG. 7-10.—Altitude control by an air port.

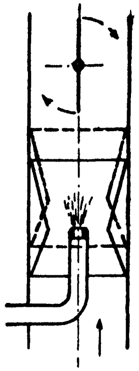

FIG. 7-11.—Altitude control by moving the venturi.

As the throttle is opened, the needle valve *n* is gradually opened and soon closes the orifice *i*. Instead of a float chamber, this carburetor uses a diaphragm pressure regulator *d*, *d*. During the operation the chambers between the diaphragm and the main fuel passageway are full of fuel and the rate of flow of the fuel regulates the position of the valves *v*, *v* actuated by the diaphragm. This arrangement prevents the interruption of

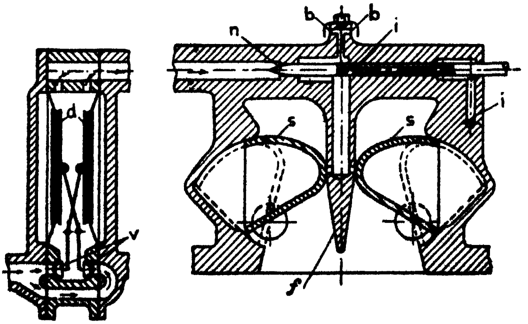

FIG. 7-12.—Aircraft carburetor with variable fuel orifice.

fuel flow due to splashing, when the aircraft is undergoing various maneuvers: sharp banking, diving, or inverted flight.

Figure 7-13 illustrates the fourth method, used in a Stromberg carburetor: A connection controlled by valve *a* between the throat of the venturi and the top of the float chamber creates a partial vacuum above the fuel and thus reduces its flow. Valve *b* in the air vent helps to adjust the desired pressure.

Ice Formation.—The evaporation of the fuel in the venturi and intake manifold requires the addition of heat to the fuel. This heat is taken from anything with which the evaporating fuel comes in contact, chiefly from the incoming air. As a result the temperature of the air drops below the dew point of water vapor, and the latter, always contained in the air, may condense and freeze into ice on the throttle plate. A considerable accumulation of ice throttles the engine and may be disastrous.

Tests have shown that appreciable ice formation takes place when the temperature of the venturi is cooled to about 9 F below the dew point of the atmosphere, if the atmospheric temperature is below 32 F.[1] Ice formation increases with rich mixtures and highly volatile gasolines.

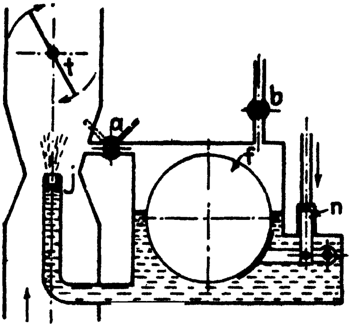

FIG. 7-13.—Float-chamber altitude control.

Heating of the intake air or the introduction of alcohol, which mixes with the water vapor and lowers its freezing point, will reduce and even eliminate the dangerous formation of ice in the induction system.

Fuel Injection.—Lately some aircraft spark-ignition engines have been using gasoline injection instead of carburetors. According to this method, the fuel is injected in a finely atomized state either into the intake manifold or directly into the cylinders. The arrangement has some decided advantages:

(*a*) It eliminates the danger of ice-formation in the induction system, because vaporization takes place in a region where the temperatures are high;

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[1] ALLEN, H. H., ROGERS, G. C., and BROOKS, D. C., Ice Formation in Aircraft Engine Carburetors, *SAE Journal*, vol. 35, No. 5, p. 417, 1934.

(*b*) gives easy control of the desirable air-fuel ratio;

(*c*) is particularly suitable for supercharging and in this case permits the attaining of an increased charge efficiency;

(*d*) has no float-chamber troubles which are caused by violent aircraft maneuvers;

(*e*) eliminates the danger of backfiring.

Against these advantages, however, it has some disadvantages: It involves considerably more parts, is heavier, and costs more to manufacture and maintain.

So far this system has not gained much favor in this country.[1]

7-5. Accessories of Carburetion. *Heating.*—It takes considerably more heat to vaporize 1 lb gasoline than its latent heat of vaporization, because of the high temperature necessary for complete evaporation of the heavier fractions. In order not to affect the rate of fuel delivery from the carburetor, it is better not to change its temperature directly or indirectly and to heat the mixture after it leaves the carburetor.

The heat necessary for the evaporation of gasoline comes from the following sources:

(*a*) heat content of the incoming air, in automobile engines preheated by the radiator and by radiation from the exhaust manifold;

(*b*) surfaces in the intake manifold heated by exhaust gases;

(*c*) mixing with hot residual gases in the engine cylinder;

(*d*) contact with hot walls of the cylinder and piston head;

(*e*) compression of the charge.

The quantity of heat and the temperature at which it must be added depend upon fuel characteristics, air-fuel ratio of the mixture, and the part of evaporation required.

The amount of heat obtained from the exhaust gases, in the manifold, can be found from the general expression

$$Q = UA(t_2 - t_1)\tau \qquad (7\text{-}13)$$

where U is the heat-transfer coefficient, Btu/sq ft-°F-hr,
A is the area of surface transmitting the heat, sq ft,
t_1 is the average temperature of the mixture, °F,
t_2 is the average temperature of the exhaust gases, °F,
τ is time, hr.

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[1] HELDT, P. M., Fuel Injection for Aircraft Engines, *Automotive Ind.*, vol. 85, No. 6, pp. 18–21, 1941.

Coefficient U from gas to gas with a metal wall separating them will be about 5 to 15 Btu/sq ft-°F-hr, depending on the gas velocity. The higher limit can be approached if the cold side of the wall has a film of fuel which is being evaporated.

The temperature of the exhaust gases varies with the power developed and the engine speed within a rather wide range,

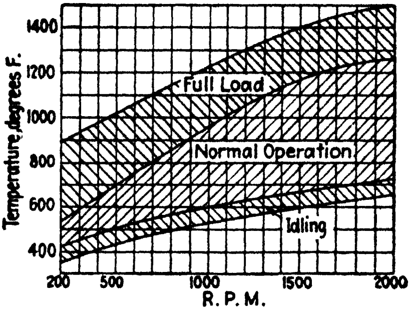

FIG. 7-14.—Exhaust temperature ranges for different loads and speeds.

Fig. 7-14. The highest relative heat requirement is near the full load at low speeds when the throttle is almost wide open, but the exhaust temperature is comparatively low.

The unvaporized fuel collects at the inner corner of T-connections or right-angle turns, and that is the best place to apply the heat by means of a jacket, Fig. 7-15.

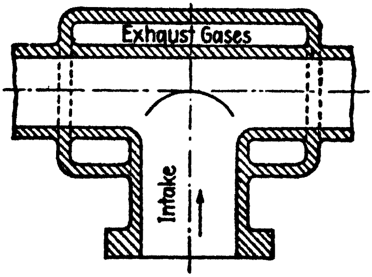

FIG. 7-15.—Heat jacket on a T-connection.

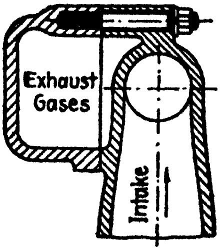

FIG. 7-16.—Hot spot in intake manifold.

The inlet and exhaust manifolds are sometimes made with a common wall at a point where the incoming mixture makes a sharp turn in order to vaporize the fuel particles which separate out there, because of inertia. This is called a *hot-spot* vaporizing system, Fig. 7-16.

Experience shows that in hot-spot manifolds the area of the heated surface should be 1 sq in. per 30 cu in. piston displacement if the surface is near the exhaust ports, increasing the area up to 1 sq in. per 15 cu in. displacement if the distance from the nearest exhaust port is 12 in. or more.

Example 7-5.—Determine the heating surface required theoretically for a 3-in. × 4-in. six-cylinder engine. Assume a charge efficiency of 0.87 at 360 rpm and wide open throttle; outside temperature 50 F.

Piston displacement is $V = 0.7854 \times 3^2 \times 4 \times 6 = 169.6$ cu in.

Charge drawn in is $169.6 \times 360 \times 60 \times 0.87/(2 \times 1728) = 922$ cu ft/hr

Assuming a gasoline discussed in Example 3-5, for which the dew point at 100 per cent evaporated and $r_{a/f} = 15$ was found as 157 F, the volume of 1 lb gasoline and 15 lb air can be computed as

$$\frac{379 \times 617}{520 \times 114} + \frac{15 \times 379 \times 617}{520 \times 29} = 4.0 + 232.7 = 236.7 \text{ cu ft}$$

The weight of gasoline per hour is $922/236.7 = 3.90$ lb/hr.

Heat necessary for the evaporation and heating of the air was found in Example 3-5 as 612 Btu/lb fuel, or a total of $612 \times 3.90 = 2390$ Btu/hr.

Thus using expression (7-13), $U = 10$, and $t_2 = 900$ F from Fig. 7-14, since 360 rpm is about one-tenth of the full speed,

$$A = \frac{Q}{U(t_2 - t_1)} = \frac{2390}{10(900 - 157)} = 0.322 \text{ sq ft or } 46 \text{ sq in.}$$

With $U = 15$, the area would be $46 \times 10/15 = 31$ sq in.

Referring to the piston displacement, this gives 1 sq in. for every 3.6 to 5.5 cu in., or twice to four times more than actual experience shows. The discrepancy is explained by the fact that the air comes to the carburetor not with a temperature of 50 F but preheated by the radiator and the exhaust manifold, and also that the evaporation is not completed in the intake manifold.

Example 7-6.—Find the heating area, using data of the previous example, for 50 per cent evaporation.

From Example 3-4 the dew point will be 88 F and the necessary heat 189.1 Btu/lb; therefore

$$A = \frac{3.88 \times 189.1}{10(900 - 88)} = 0.0904 \text{ sq ft or } 13 \text{ sq in.}$$

which corresponds to 1 sq in. for 13 cu in. displacement—almost accurate.

Heat Regulation.—As already mentioned, the heat requirements change with the load and speed of the engine and with weather conditions. Excessive heat is undesirable because it reduces the weight of the charge.

In some automobile engines the exhaust gases from all cylinders are used in the intake-manifold heater during the warming-up period, and a special valve by-passes the greater part of the exhaust gases after this period.

To avoid unnecessary heating of the charge during warm weather, many automobile engines have a special valve in the exhaust manifold which in a certain position allows the exhaust gases to escape without heating the intake manifold.

Some engines have in the heating system a valve which may be adjusted to change gradually the amount of exhaust gases admitted to the heater, depending on weather conditions. This valve is connected with the throttle mechanism and decreases the heat rapidly with opening of the throttle.

Intake manifolds must be designed to ensure equal distribution of the charge to each cylinder. Unequal distribution may be caused by different resistances and sharp turns in the connections between the carburetor and cylinder and also by the inertia of the high-velocity current which may have a tendency to increase the flow to certain cylinders at the expense of others and also to separate suspended fuel. The inertia influence varies with the engine speed and valve timing.

To prevent an excessive increase of the resistance, thereby giving a lower charge efficiency, the air velocity should not exceed certain limits. These limits, at 1000 rpm, are:

Riser, updraft	90 fps
Riser, downdraft	80 fps
Horizontal manifold branches	65 fps

At the same time the area of the inlet-manifold cross section should not be made too large, as some of the particles of gasoline, not vaporized but only kept in suspension, may drop out if the velocity is below 2400 fpm.

Figure 7-17 gives the numerical relation between engine displacement, rotary speed, size of the intake manifold, and manifold gas velocity. The diagram is computed for useful displacement, meaning displacement multiplied by an average charge efficiency, $\eta_{ch} = 0.85$. The example in Fig. 7-17 shows that a 320 cu in. engine at 2800 rpm requires a $1\frac{3}{4}$-in. manifold, which gives a good average gas velocity of 13,000 fpm; the 2-in. manifold gives about 10,000 fpm, not quite enough; and the 1½-in. manifold gives about 18,000 fpm, which is too high.

The actual fitness of a manifold for a certain engine can be determined only by tests.

To avoid interference of suction from different cylinders whose intake valves are opened simultaneously, eight-cylinder engines are sometimes equipped with two carburetors and dual manifolds.

Governing.—Carburetor engines are governed by throttling. In stationary engines the desired speed is maintained by a governor acting upon a butterfly valve, and in automotive engines the desired speed is obtained by a foot or manual control of the throttle.

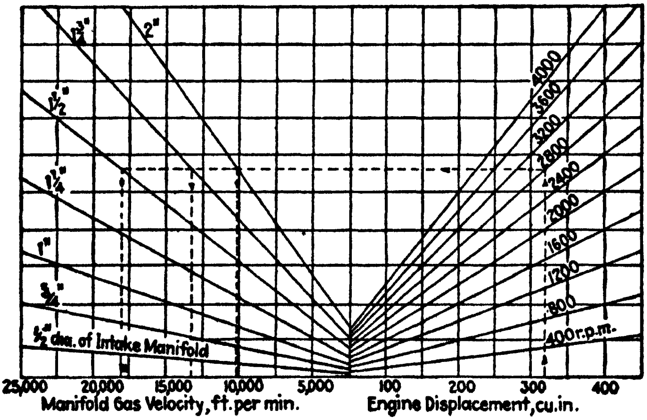

FIG. 7-17.—Intake-manifold gas velocities.

Fuel-supply Methods.—Gasoline from the storage tank may be delivered to the carburetor by gravity, vacuum, or pressure.

In the gravity-feed system, as used in some automobiles, the gasoline tank is above the carburetor. The elevation required depends upon the size and length of the tube connecting the fuel tank and the carburetor and usually varies from 12 to 18 in.

In the vacuum system a small tank is placed above and near the carburetor. A vacuum created in the tank through a connection from the intake manifold pulls the fuel from the storage tank. The vacuum tank has a float which shuts off the connection to the intake manifold when the fuel reaches a certain level and opens a connection to the air. From the vacuum tank the fuel flows by gravity to the carburetor. When the tank is emptied, the float operates certain valves so as to refill it. Thus the operation of the system is intermittent.

Pressure systems may be of two different types. In the older system a small pressure in the storage tank forced the gasoline to the carburetor. The air was pumped by a small pump driven from the engine. A pressure-relief valve, a pressure gage, and a hand pump for starting were also necessary for this system.

In the modern pressure system a mechanically or electrically operated diaphragm pump is placed between the storage tank and the carburetor.

Vapor Lock.—If the supply tube is small and the amount of gasoline drawn is large, a high velocity will result. This will cause a considerable pressure drop in the line and a vacuum, which may cause the formation of vapor bubbles. If these bubbles accumulate in the highest point of a tube bend, they may interrupt the gasoline flow to the vacuum tank or fuel pump, and lack of fuel will stop the engine. Vapor lock is caused by the presence of low-boiling fractions in a gasoline and usually happens on a hot day.

In an automobile engine, vapor lock occurs when the engine runs in low gear with a wide-open throttle, as when climbing a steep grade. In an airplane engine it may happen during the take-off when the engine also operates with wide-open throttle.

7-6. Problems.—1. Compare the weights of air flowing through a venturi when found from expressions (7-5) and (7-7) for intake pressures of 14.7, 14.0, and 13.7 psi and for a depression of 1 psi in the throat.

2. Determine the size of the fuel jet of a simple carburetor for a six-cylinder, four-stroke engine with a $3\frac{1}{4}$-in. bore and a $4\frac{1}{4}$-in. stroke, normal speed 3000 rpm. Diameter of the venturi throat is $1\frac{1}{4}$ in. Air-fuel ratio desired is 15:1. Assume a volumetric efficiency of 85 per cent.

3. Determine the air-fuel ratio if with a decrease of the load in problem 2 the throttle is partially closed, decreasing the vacuum in the throat by 10 in. of water column.

4. Determine the air-fuel ratio if with an increase of the load in problem 2 the throttle is opened, giving a depression of 50 in. water; use a jet diameter of 0.063 in.

5. (*a*) Determine the size of the fuel jet of a simple carburetor for a four-cylinder four-stroke $3\frac{1}{16}$-in. × $4\frac{1}{4}$ in. engine, normal speed 3200 rpm. Select the diameter of the venturi throat so that the air velocity is not too

high. The desired air-fuel ratio is 14.9:1; assume a volumetric efficiency of 87 per cent. (*b*) Find the venturi depression.

6. (*a*) Determine the velocity of the fuel discharge from a jet 0.050 in. in diameter; the venturi throat diameter is 1 in., the depression is 30 in. of water; the fuel has a specific gravity of 0.71. Find (*b*) the air velocity and (*c*) the air-fuel ratio.

7. Work problem 6 with a depression of 40 in. due to an increase of the load.

8. Taking values of the depression in the venturi throat 20, 40, and 60 in. water for the carburetor in problem 2, plot a curve of air-fuel ratios, assuming constant coefficients of discharge.

9. Determine the change of air-fuel ratio in an airplane-engine carburetor at altitudes of 5000, 10,000, and 18,000 ft. Carburetor is adjusted for 15:1 ratio at sea level at standard pressure and 60 F.

10. A simple carburetor is adjusted for a 15.2:1 air-fuel ratio at sea level and 60 F. Determine the air-fuel ratios at 10,000 ft, 16,000 ft, and 20,000 ft altitudes.

11. Determine the heating surface theoretically required for the engine in problem 2, with an outside temperature of 40 F. Use gasoline and r_{af} as discussed in Example 3-5.

12. Determine the heating surface in problem 11 from practical data as a function of the piston displacement.

13. Determine the heating surface theoretically required for the engine in problem 5 with an outside temperature of 42 F, using gasoline discussed in Example 3-5 and assuming 90 per cent evaporation.

14. Determine the heating surface in problem 13 from practical data, as a function of the piston displacement.

15. Using Fig. 7-17 find the manifold size for the engine in problem 2.

16. Using Fig. 7-17 find the manifold size for the engine in problem 5.

17. Compute the minimum diameters in the downdraft riser and the horizontal branches of the intake manifold of the engine of problems 2 and 15; also check whether the venturi throat is not too small.

18. Compute the minimum diameters in the updraft riser and the horizontal branches of the intake manifold for the engine of problems 5 and 16.

CHAPTER 8

COMBUSTION IN SPARK-IGNITION ENGINES

8-1. Ignition. *Ignition Temperature.*—The temperature at which a fuel will start to burn under given conditions is called its *ignition temperature.* This temperature varies greatly with the composition of the air-fuel mixture, amount of excess air, dilution with burned gases, with the pressure of the mixture, with the size, shape, and temperature of the compression space, and with the method of raising the temperature until ignition occurs. Of all these factors one of the most important is the pressure of the gas, as shown by Fig. 14-1.

However, the ignition temperature appears to be influenced not so much by the pressure as by heat transfer, which itself increases with the density of the air-fuel mixture. The ignition temperature T, degrees Rankine, can be expressed by the empirical relation

$$T = c/w^m \tag{8-1}$$

where c and m are constants and w is the specific weight of the charge when it ignites, pounds per cubic foot. For aliphatic hydrocarbons, C_nH_{2n+2} and C_nH_{2n}, these constants can be taken as $c = 820$ and $m = 0.16$; for aromatic fuels, C_nH_{2n-6}, they vary, for c, from 1030 to 1100 and, for m, from 0.193 to 0.23.

Igniters.—In the great majority of present low-compression engines, ignition is effected by a high-tension spark which is produced by the breaking of an electric current that passes between the points of a spark plug. This current is furnished either by a storage battery connected to an induction coil or by an oscillating or rotating magneto.

In most oil engines with moderate compression, the fuel is injected considerably before top dead center. The air-fuel charge is ignited by a hot or incandescent piece in the cylinder head, by the walls of a bulb-shaped compression space, or by a tongue or projection in the compression space. These engines are called ***hot-bulb*** or ***hot-surface-ignition*** engines. In some oil engines with moderate compression the fuel is injected as a finely atomized spray, which is ignited by a spark plug, as in a carburetor engine.

The temperature of the hot spot causing ignition must be considerably higher than the ignition temperature of the fuel, some 150 to 350° in the case of gasoline which ignites at 650 F.[1]

8-2. Combustion. *Flame Propagation.*—When a combustible mixture is ignited, two different phenomena occur—slow combustion and a compression wave. In the slow combustion the fuel molecules that are already burning raise by conduction and radiation the temperature of adjacent molecules, causing them to ignite; at the same time the temperature rise of the gas molecules increases their velocity; this raises the pressure in that point and results in an expansion that assists in propagating the ignition. Turbulences created in the charge before ignition increase materially the velocity of flame propagation, which in this case occurs not only through conduction and radiation but also through convection.

While it is not easy to segregate these processes, experiments show that in a quiet mixture the velocity of slow or conduction ignition may vary between 2 and 50 fps, depending upon the composition, temperature, and pressure of the charge, shape of combustion chamber, location and action of the igniter, and, in the case of an electric spark plug, upon the heat of the spark.

If a compression wave occurs, its velocity may reach several thousand feet per second, and passing through the explosive mixture accompanied by almost instantaneous generation of very high pressures it may cause serious trouble in the engine.

Experiments on the actual velocity of flame propagation in internal-combustion-engine cylinders give in the absence of excessive compression waves values ranging between 30 and 125, with an average of about 70 fps,[2] chiefly depending upon the character of turbulence, which itself depends to a great extent on the engine speed and improves with an increase of speed. It may be assumed that the flame velocity c_t increases from about 30 fps at low speeds of 300 rpm and below up to 125 fps at 3600 rpm. Thus, if under certain conditions combustion lasts a certain time interval, which in turn corresponds to a certain crank travel, and if the engine speed is increased, combustion usually is completed in a shorter time interval on account of improved turbulence and increased velocity of flame propagation and corresponds to approximately the same crank travel.

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[1] *NACA Tech. Mem.* 873, 1938.

[2] Marvin, C. F., Jr., and Best, R. D., *NACA Rept.* 399, 1931.

It must be noted that the flame front does not have a smooth shape but is irregular.[1] Furthermore, burning is not completed in a shallow reaction zone, but the reaction and heat liberation continue for some time after the flame front has passed through the charge.[2] The velocity of flame propagation is influenced greatly by the air-fuel ratio and reaches its maximum at 85 to 90 per cent of theoretical air; below and above this amount the velocity decreases, and with a theoretical air it is about 10 per cent lower. An increase of the compression ratio increases the flame velocity only slightly, and the presence of tetraethyl lead does not influence the reaction speed.[3]

Explosiveness.—In order that a mixture of combustible gas or vapor with air shall burn so fast that the combustion is called an explosion, the ratio of air to gas must lie within certain limits, depending on the characteristics of the fuel. In general a mixture of a fuel with the theoretically necessary air gives the highest pressures and temperatures, but the highest speed of flame propagation is obtained with 85 to 90 per cent of theoretical air. If the air-fuel ratio is either increased above or decreased below the corresponding value, the flame-propagation velocity is decreased, until at certain upper and lower limits the mixture is no longer explosive and only slow combustion can take place.

Table 8-1 gives the limits of explosiveness expressed in air-fuel ratios. The data should be considered as only approximate. The explosiveness in engine cylinders may be considerably different from values of Table 8-1, as many conditions affect them—turbulence, pressure, temperature, and particularly dilution with inert gases.

For spark-ignition engines the maximum air-fuel ratios are of particular interest, as they show how lean a mixture can be made before it ceases to be explosive and usable in this type of engine.

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[1] RATHROCK, A. M., and SPENCER, R. C., *NACA Tech. Notes* 603, 1937.

[2] FIOCK, E. F., MARVIN, C. F., JR., CALDWELL, F. R., and ROEDER, C. H., *NACA Rept.* 682, 1940.

[3] *NACA Tech. Mem.* 668, 1932.

Temperature.—The temperature of the gas in the combustion space varies during combustion and is not uniform, as can be seen from the analysis of the combustion process. As a result, after combustion is practically completed, there exists a difference of 600 to 1000 F., depending upon turbulence, velocity of flame propagation, etc., between the temperatures of the parts burned first and last.[1] Measurements of maximum temperatures in the combustion space of engines must therefore be considered as giving maximum temperatures of the flame rather than of the gas. The two factors most influencing these temperatures are air-fuel ratio and spark timing. Figure 8-1 shows the influence

TABLE 8-1.—LIMITS OF EXPLOSIVENESS
($p = 14.7$ psia; $t = 60$ F)

Fuel	Air-fuel ratio by weight			Fuel	Air-fuel ratio by weight		
	Theoret. for complete combustion	Exploding			Theoret. for complete combustion	Exploding	
		Min.	Max.			Min.	Max.
Hydrogen	34.6	5	340	Carbon monoxide	2.5	0.3	7.3
Hydrogen sulphide	6.1	1	9	Benzene	13.3	4	26
Methane	17.3	11	32	Hexane	15.3	5	27
Acetylene	13.3	1	43	Toluene	13.5	4	24
Ethylene	14.9	4	30	Gasoline	15.2	4	19
Ethane	16.1	6	30	Methyl alcohol	6.5	2	14
Propane	15.7	6	27	Ethyl alcohol	9.0	3	18

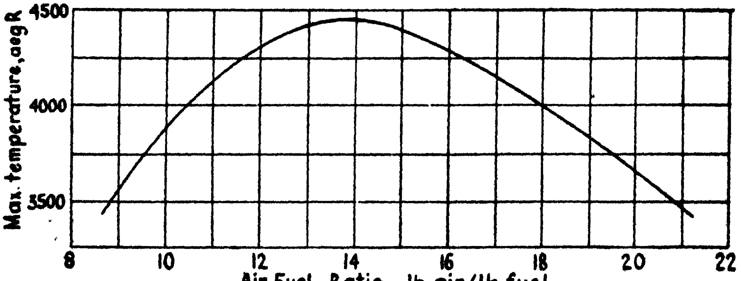

FIG. 8-1.—Influence of air-fuel ratio on maximum temperature.

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[1] TAYLOR, E. S., *NACA Tech. Notes* 533, 1935.

of air-fuel ratio for a constant ignition timing in a gasoline engine with a compression ratio of 3.86:1, and Fig. 8-2 shows the influence of the ignition timing and also of the air-fuel ratio found by actual measurements.[1] Thus the highest temperature in the combustion space may reach about 4400 F. However, normally the highest temperature is considerably lower, as may be seen from Fig. 12-2.

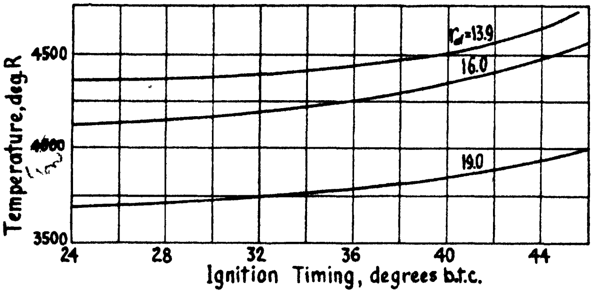

FIG. 8-2.—Influence of ignition timing and air-fuel ratio on maximum temperature.

8-3. Detonation.—Detonation, also called *fuel knocking* or *pinging*, presents an example of a phenomenon which was known to the practical engineer a long time before the theory behind it was established and its far-reaching importance realized. Detonation is the most important of all factors that limit the power output and efficiency of engines using volatile liquid fuel and spark ignition. Detonation is influenced to a great extent by the engine design, but essentially it is a chemical and physical problem.

The noise which has given the phenomenon the name knocking or pinging is caused by a blow delivered against the cylinder wall by a high-pressure wave traveling at a very great speed, 3000 to 9000 fps, through the gas. Detonation knock resembles, to a certain extent, preignition knock, which is caused by overheated spark-plug points or scraps of stray incandescent carbon. However, preignition gradually turns into violent thumping, and the excessive pressures developed before the end of the compression stroke reduce the power output of an engine and soon bring it to a stop. Detonation, on the other hand, develops after ignition has been started by the spark; if the detonation

[1] HERSHEY, R. L., and PATON, C. R., *Univ. Illinois Bull.* 262, 1933.

is not violent, it may persist for long periods without noticeably affecting the engine performance. If detonation increases, it may cause loss of power; and if it becomes very violent, it is apt to cause the overheating of the spark-plug points, thus starting preignition and stopping the engine altogether.

The two phenomena are totally distinct from one another: preignition precedes the spark; detonation follows it. While preignition may occur with any fuel, detonation occurs only with volatile liquid fuels and depends upon their composition and molecular structure, other contributing factors being high pressure, high temperature, and an insufficient heat transfer to the combustion-chamber walls.

There are several theories dealing with the causes of detonation. One theory holds that the high temperature of the products formed in the flame front results in their expansion, which compresses the unburned part of the air-fuel mixture and raises its temperature until self-ignition occurs ahead of the flame front.[1] According to a newer chemical theory, slow combustion begins during the compression stroke at about 320 F and forms unstable compounds, peroxides, of which $CH_3(OH)$ is the simplest one;[2] after regular ignition has started, these unstable compounds, compressed by the combustion, fall apart and ignite so violently that they produce a heavy pressure wave striking the cylinder walls.

There are also several theories that explain the loss of power that accompanies detonation. According to the first theory, the high temperatures created during detonation cause an increased heat loss through radiation, whereas, according to the second theory, the power loss is caused by a mass vibration transformed into heat, which is lost subsequently to the combustion-chamber walls.[3]

Detonation Factors.—The factors that influence detonation are:

1. Fuel characteristics:
    (*a*) molecular structure,
    (*b*) temperature of self-ignition,
    (*c*) rate of burning.

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[1] POPE, A. W., JR., and MURDOCK, J. A., *SAE Journal*, vol. 30, No. 3, p. 136, March, 1932.

[2] DEVILLERS, "Le Moteur d'explosion," p. 89, Dunod, Paris, 1935.

[3] GOOD, C. W., *Trans. ASME*, vol. 64, No. 4, p. 320, May, 1942.

2. Conditions of the cylinder charge:
   (*a*) air-fuel ratio,
   (*b*) charge distribution,
   (*c*) temperature of the charge,
   (*d*) density of the charge.
3. Compression ratio:
   (*a*) compression pressure,
   (*b*) compression temperature,
   (*c*) dilution by residual gases,
   (*d*) valve timing.
4. Ignition:
   (*a*) location of spark plugs,
   (*b*) timing.
5. Combustion chamber:
   (*a*) shape,
   (*b*) material,
   (*c*) surface conditions.

Fuel Characteristics. *Molecular Structure.*—In the order of decreasing tendency to detonate, the fuels are: paraffins, naphthenes, and aromatics. In general, it appears that for most hydrocarbons a more compact molecular structure is associated with a lower tendency to detonate.

Ignition Temperature.—While there is no strict relation between self-ignition temperatures and detonation, in general fuels with higher self-ignition temperatures are less detonating. However, the addition of antiknock substances does not influence the temperature of self-ignition.

Rate of Burning.—A high rate of burning allows less time for the unburned portion to lose heat to the combustion-chamber walls, which tends to promote detonation. On the other hand, a high rate of burning allows less time for the unburned portion to attain the temperature of self-ignition, which tends to decrease detonation. Finally, a high rate of burning requires a smaller spark advance, which results in the burning of the last portion of the charge nearer the dead-center piston position and tends to decrease detonation. Thus the effect of the rate of burning is rather involved. However, in general the detonating tendency decreases with a decrease of the rate of burning.[1]

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[1] Ricardo, H. R., and Glyde, H. S., "The High-speed Internal-combustion Engine," p. 30, Interscience Publisher, Inc., New York, 1941.

Conditions of Cylinder Charge. *Air-fuel Ratio.*—Detonation seems to be worst at the condition of maximum power and highest flame-propagation speed at around 85 per cent of theoretical air or an air-fuel ratio of about 13:1. Both an increase and a decrease of the air-fuel ratio as compared with this value lower the knock intensity.[1]

Charge Distribution.—The different fractions of a commercial gasoline have different detonating characteristics, Fig. 8-3.[2] The more volatile fractions have higher octane numbers and are completely evaporated in the manifold and therefore distributed more or less uniformly to all the cylinders of a multicylinder engine; the less volatile fractions are in a liquid form and are therefore less evenly distributed. This results in appreciably different knocking tendencies in the various cylinders. The same difference in knocking tendency exists in a multicylinder engine when individual cylinders do not receive the same charge due to improper manifolding.

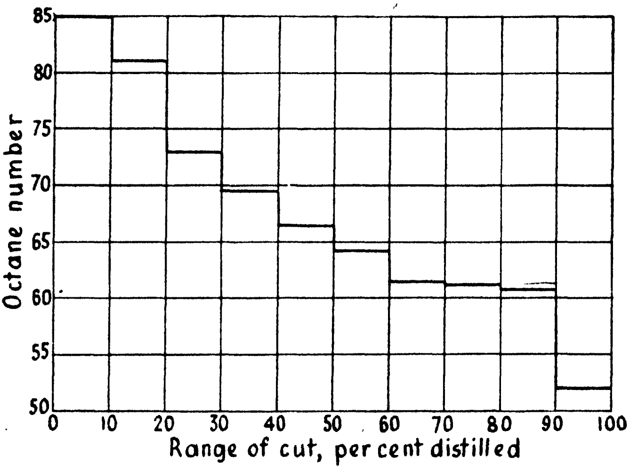

FIG. 8-3.—Octane numbers of different cuts of distilled gasoline.

By making the carburetor mixture richer the knocking can be reduced to an incipient, harmless detonation in those cylinders that have the greatest tendency to knock. Obviously, such adjustments will decrease the thermal efficiency and may also reduce appreciably the power output of the engine.

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[1] TAYLOR, C. F., and TAYLOR, E. S., "Internal-combustion Engines," p. 97, International Textbook Company, Scranton, Pa., 1938.

[2] BARTHOLOMEW, E., CHALK, H., and BREWSTER, B., Carburetion, Manifolding, and Fuel Anti-knock Value, *SAE Journal*, vol. 42, pp. 141, 148, 1938

Charge Temperature.—An increase in the mixture temperature and the resulting increase of the charge temperature increase the evaporation of the fuel in the intake manifold and thus give a better distribution of the fuel supplied or of the heavier fuel fractions. As explained above, this reduces the tendency to detonate. Also, the density of the charge is reduced with an increase of the mixture temperature, and together with it the combustion pressure decreases. This also reduces the tendency to detonate. On the other hand, an increase in the charge temperature raises the temperature of the unburned portion of the charge, which increases the tendency to detonate. This latter unfavorable influence is stronger than the first two favorable effects. It can be offset by lowering the air-fuel ratio, but with an increase of the specific fuel consumption.[1]

The lowering of the gas temperature during the compression and burning through a lowering of the water-jacket temperature and cooling of exhaust valves has a beneficial influence in decreasing the tendency to knock.

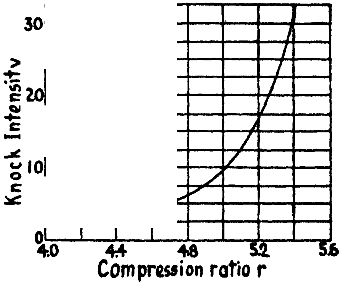

Fig. 8-4.—Effect of compression ratio on detonation.

Charge Density.—Throttling of the intake reduces the tendency to detonate, whereas supercharging increases this tendency. The change of density affects the tendency to detonate, chiefly by changing the dilution by residual gases: an increased dilution retards the rate of burning, and this, as already mentioned, lowers the tendency to detonate.

Compression Ratio. *Compression Pressure.*—An increase in the compression pressure, all other factors being equal, increases the maximum pressure during combustion and therefore the tendency to knock.

Compression temperature increases with an increase in the compression ratio, and this also increases the tendency to detonate. For this reason, with a high compression ratio, exhaust valves with salt or sodium cooling must be used.

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[1] Peletier, L. A., *NACA Tech. Mem.* 853, 1938.

The influence of the compression ratio is very pronounced, Fig. 8-4.[1]

Dilution.—An increase of the compression ratio decreases dilution by residual gases and thus increases the rate of burning and hence the tendency to knock.

Valve Timing.—The opening of the intake valve before the exhaust valve is closed influences the percentage of dilution with exhaust gases. The intake pressure and, hence, the charge density, depend upon valve timing. Finally, a late closing of the intake valves lowers the actual compression ratio. All these factors have an appreciable effect on detonation.

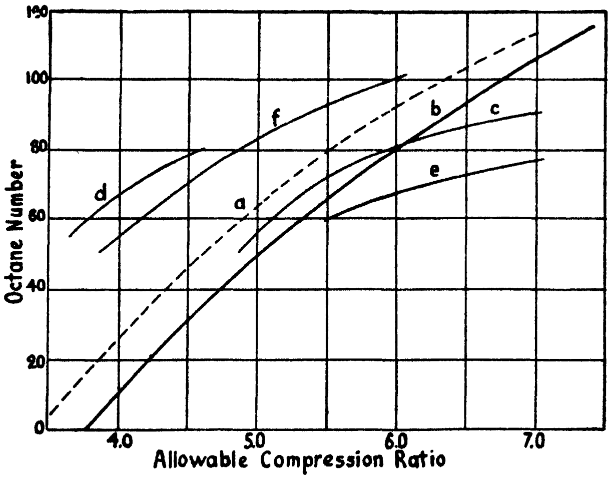

FIG. 8-5.—Influence of octane number on allowable compression ratio.

Allowable Compression Ratio.—Both the power output and thermal efficiency of an engine are increased by an increase of the compression ratio; therefore it is logical to use the highest possible compression ratio. In gas engines the limit is set usually by the danger of preignition; in engines using volatile liquid fuel, by fuel knocking. *Allowable* compression ratio is a compression ratio with an incipient, harmless detonation. The term *highest useful compression ratio*, H.U.C.R., refers to a compression ratio at which a particularly efficient fuel-test engine can be operated without detonation at any mixture strength or with any ignition tuning at a speed of 1500 rpm.[2] All other conditions being equal, the allowable compression ratio depends upon the antiknocking characteristics of the fuel; the higher the octane number, the higher can be the compression ratio. On the other hand, the tendency to detonate depends upon a combination of so many factors that the same fuel will give different allowable compression ratios in different, although similar, engines.

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[1] TAYLOR and TAYLOR, *op. cit.*, p. 97.

[2] RICARDO and GLYDE, *op. cit.*, p. 10.

It is impossible to express the relation between compression ratio and octane number by a definite formula or curve. A picture of this relation may be obtained from Fig. 8-5 in which curve *a* corresponds to data obtained from Ricardo's engine, using a mixture of pure heptane, C_7H_{18}, with benzene, C_6H_6, which has an octane number of about 110; the other curves were obtained with various single and multicylinder engines running at speeds from 600 to 2250 rpm; they all show the same general trend.

Ignition. *Spark-plug location* has a great influence upon tendency to detonate: The larger the distance from the ignition point to the farthest point of the combustion space, the longer is the time required to burn a given fraction of the charge and the greater is the time available for the compression ignition and detonation of the last portion of the charge. Therefore, all other conditions being equal, larger engines have a greater tendency to detonate than smaller engines, and their allowable compression ratio is lower. Also, if the spark plug is located near the inlet valve, the exhaust valve assists in increasing the temperature of the last portion of the charge. Detonation tendency will be decreased by locating the spark plugs near the exhaust valve, the tendency will be decreased even more by using two spark plugs, thus cutting down the distance of the flame travel to the

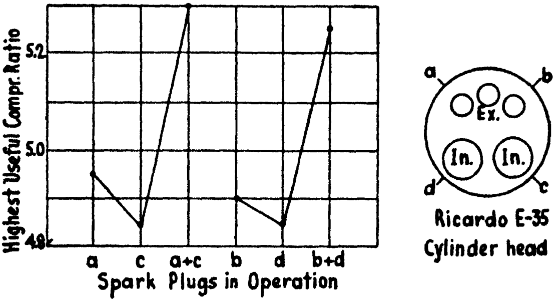

FIG. 8-6.—Effect of spark-plug location on detonation.

last portion of the charge to ignite. Figure 8-6 illustrates this in terms of Ricardo's highest useful compression ratio.[1]

Timing.—A change in ignition timing changes the amount of the fuel burned before and after the dead-center position of the piston. Thus, advancing the spark timing increases the temperature of the charge, and increasing the rate of burning results in the last portion of the fuel burning nearer to the dead center. These factors increase the tendency to detonate, Fig. 8-7.[2] This tendency can be lowered or even stopped by retarding the spark, however, with an unavoidable loss of power and thermal efficiency. The relation between the optimum spark advance and that required to avoid detonation depends on the fuel. The spark advance for incipient detonation may be smaller or greater than the optimum spark advance at any engine speed if the octane number is below or above, respectively, that required for optimum spark advance.

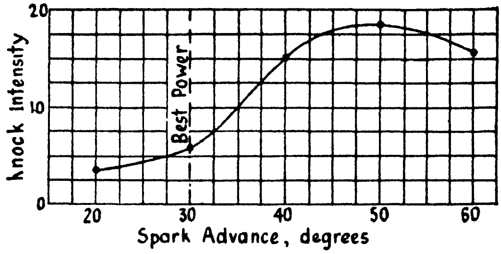

FIG. 8-7.—Effect of ignition timing on detonation.

8-4. Ignition Timing. *Spark Advance.*—In the explosion cycle the maximum useful work is obtained with a combustion at constant volume, with the maximum pressure occurring near the dead center. Since a certain time element elapses after ignition and before the maximum pressure is attained, the charge must be ignited before the dead center. However, too early ignition results in a large negative work at the end of the compression stroke and excessive combustion pressures.

Experimental data[3] indicate that for best results the spark advance should be such that half of the pressure rise occurs at dead center, which practically corresponds to 75 per cent of the explosion time after ignition. Designating by s, in., the distance from the point of ignition, the spark plug, to the farthest point of the compression space, n the engine speed, rpm, and c_i the velocity of ignition, fps, the above stated rule gives for the angle of optimum spark advance

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[1] RICARDO and GLYDE, *op. cit.*, p. 53.

[2] TAYLOR and TAYLOR, *op. cit.*, p. 98.

[3] *SAE Trans.*, vol. 18, pt. II, p. 117.

$$\alpha^0 = \frac{0.75 \times 360 \times n \times s}{60 \times 12 \times c_i} = \frac{0.375ns}{c_i} \qquad (8\text{-}2)$$

Distance s being a function of and often equal to the diameter of the cylinder, expression (8-2) shows that the angle α must be increased with increase of speed and size of engine. However, c_i itself increases with n on account of increased turbulence so that α increases more slowly than it would seem from expression (8-2). In present engines, α varies from 10 to 40°.

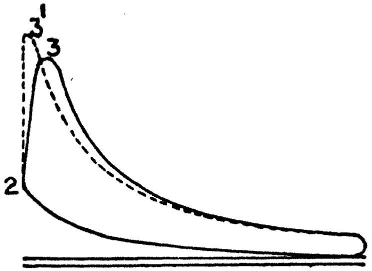

FIG. 8-8.—Influence of ignition timing.

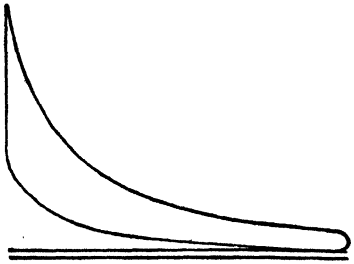

FIG. 8-9.—Early ignition.

Indicator diagrams taken from engines in operation give information regarding ignition and combustion. Figure 8-8 shows a diagram with late ignition, line 2-3. By advancing the ignition, a better combustion line 2-3′ was obtained. By further increasing the angle of spark advance, diagram Fig. 8-9 was obtained, which still nearer approaches a theoretical diagram with constant-volume combustion but is not very desirable because it shows a higher maximum pressure with no appreciable increase of thermal efficiency.

If the turbulence is not sufficient for the air-gas mixture used, or the mixture is not uniform, as is often the case in gasoline engines, consecutive diagrams taken on the same indicator card will not cover each other, Fig. 8-10. Advancing the spark may help some if the mixture is not too lean. Similar diagrams are obtained if the ignition is not functioning properly—either the intensity of the spark or its timing varies from cycle to cycle.

As the ignition and combustion events occur near the dead center when the engine piston and indicator drum are hardly moving, much better information may be obtained from a diagram, Fig. 8-11, which is taken with the indicator motion offset by 90° to the engine crank. On this diagram the events occurring near the dead centers are recorded near the middle of the card when the velocity of the indicator drum is near its maximum. An offset diagram affords opportunity to check the time of ignition and to measure the duration and velocity of combustion. Offset diagrams taken on the same card may show a scattering of consecutive cycles when an ordinary indicator diagram does not show it.

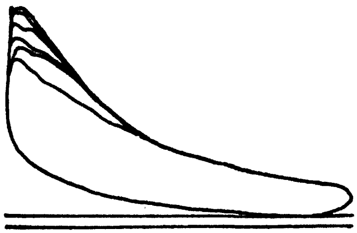

FIG. 8-10.—Scattered diagrams.

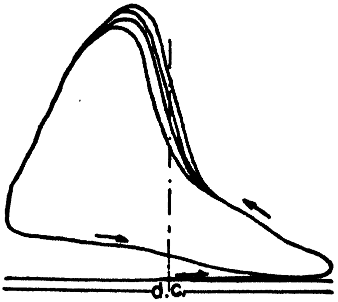

FIG. 8-11.—Offset indicator diagram.

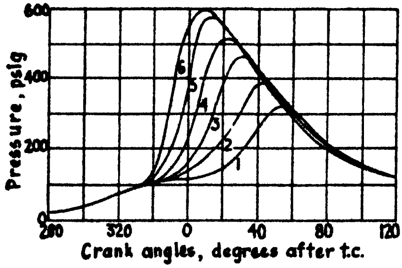

FIG. 8-12.—Effect of spark timing on maximum pressure.

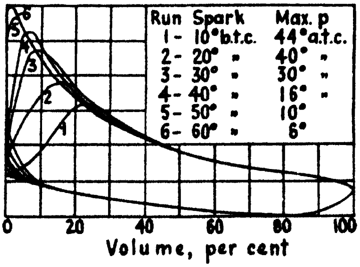

FIG. 8-13.—Effect of spark timing on pressure-volume diagram.

Still better information is obtained from a diagram taken with a uniformly revolving indicator drum in which the abscissas are proportional to the angles of the crank travel, Fig. 8-12; Fig. 8-13 gives the regular indicator diagram for the same six spark advances. The influence of the ignition timing on the power developed can be seen on Fig. 8-14, which shows that the optimum spark angle is about 35° before top center and that a retarding by about 15° lowers the power about as much as an excessive advance of 25°.

Combustion Time.—Ideal efficiencies, Fig. 5-10, were based upon instantaneous combustion at dead center. If instantaneous combustion occurs after dead center, when the crank has traveled a degrees, Fig. 8-15, the expansion ratio $r_2 = V/V_2$ is less than the ratio $r_1 = V/V_c$, and the efficiency of the cycle is lowered. If the combustion occurs before dead center, $a' = a$, the burned charge will be first compressed and then expanded and the useful expansion will again begin only from point 2 and again the efficiency will be lowered.

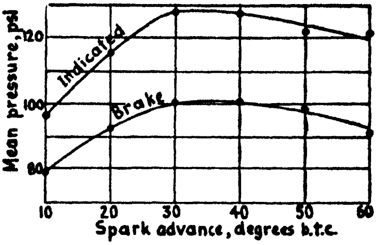

FIG. 8-14.—Effect of spark advance on mean effective pressures.

As combustion requires a certain time, different portions of the fuel will be burned at different efficiencies, but all efficiencies will be lower than if the fuel were burned at dead cehter. Figure 8-16 gives a curve of mass of fuel burned against combustion time,[1] typical for average gasoline. It shows that combustion at first goes very slowly, and it takes about 75 per cent of the total time to burn the first half of the fuel. By taking the slope of the mass-burned curve, the burning-rate curve is found. If the ignition point and combustion time are known, the average efficiency can be computed. First the piston displacements x from the dead center must be found either graphically, similar to

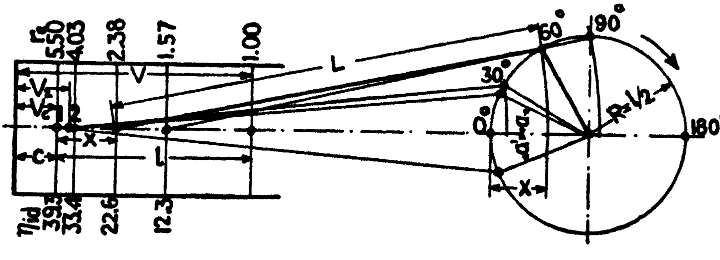

FIG. 8-15.—Expansion ratios and efficiencies for different crank angles.

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[1] MARVIN, C. F., JR., *NACA Tech. Rept.* 276, 1927.

Fig. 8-15, or from the approximate expression

$$x = R(1 - \cos a \pm R \sin^2 a/2L) \tag{8-3}$$

where R is the crank radius equal to half of the engine stroke l, L is the length of the connecting rod and a the angle of the crank with the cylinder center line, the plus sign being used for the displacements from the head-end dead center and the minus sign for those from the crank-end dead center. Using these values of x, ratios of expansion are computed from expression

$$r_e = (l + c)/(c + x) \tag{8-4}$$

where c is the relative value of the compression space. At dead center, r_e is evidently equal to $(l + c)/c$. Knowing the percentage of air, corresponding values of η_{id} are found from Fig. 5-10 and plotted against angles of crank travel. After that the burning-rate curve is plotted, starting at the ignition point and extending it over the duration of combustion. The average ideal efficiency is found now by adding the weighted mean efficiencies for small crank travels. A weighted efficiency means the efficiency multiplied by the ratio of the corresponding area under the burning-rate curve to the total area.

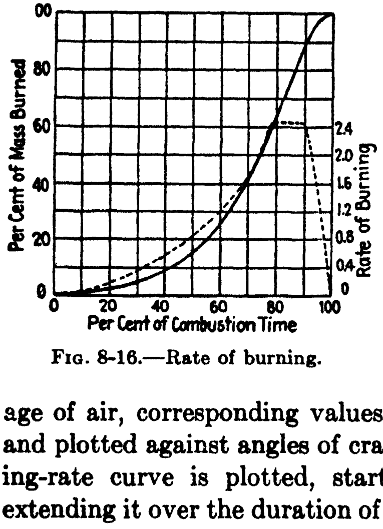

FIG. 8-16.—Rate of burning.

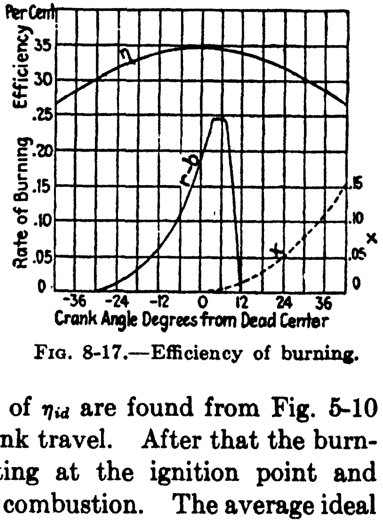

FIG. 8-17.—Efficiency of burning.

A mass-burned curve for an engine can be determined with sufficient accuracy from its pressure-crank-travel or pressure-volume diagram.[1]

Example 8-1.—Determine the average ideal efficiency for an engine with ignition 30° before dead center, duration of combustion 42°, compression

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[1] RASSWEILER, G. M., and WITHROW, L., *SAE Journal*, vol. 42, pp. 185-204, 1938.

$r = 5.5$, rate of burning as given in Fig. 8-16, amount of air 110 per cent, $L/R = 4.5$.

The burning-rate curve from Fig. 8-16 is plotted against the crank angle, Fig. 8-17; next are found the x-points from expression (8-3), using $R = 0.5$, for every 6° beginning with $-(30 + 24)/2 = -27°$ and plotted in the right half of Fig. 8-17.

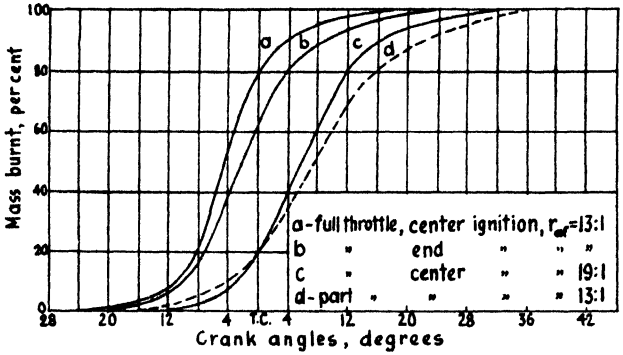

FIG. 8-18.—Effect of engine operating conditions on rate of burning.

From expression (8-4) are computed the ratios of expansion, and from Fig. 5-10 are found the corresponding values of η_{id} and plotted in Fig. 8-17. Now the areas under the r-b curve are determined for every 6°. The remaining computations are clear from the following tabulation:

Crank travel, deg. from dead center	Area under r-b curve, sq in.	Distances x, in.	Average compression ratio r	Average efficiency, per cent	Weighted efficiency, per cent
−30 to −24	0.001	0.0659	4.24	30.5	30.5 × 0.005 = 0.15
−24 to −18	0.005	0.0404	4.65	32.2	32.2 × 0.027 = 0.87
−18 to −12	0.012	0.0208	5.03	33.4	33.4 × 0.065 = 2.17
−12 to − 6	0.022	0.0075	5.33	34.0	34.0 × 0.120 = 4.09
− 6 to 0	0.040	0.0009	5.49	34.9	34.9 × 0.217 = 7.57
0 to + 6	0.063	0.0009	5.49	34.9	34.9 × 0.342 = 11.94
6 to +12	0.041	0.0075	5.33	34.0	34.0 × 0.222 = 7.53
Total........	0.184		Average ideal efficiency = 34.32		

The average ideal efficiency of 34.32 per cent found for the given conditions is close to the 34.9 per cent efficiency for instantaneous combustion at dead center, indicating that the spark advance of 30° is near the optimum one

While the general shape of the mass-burned and rate of burning curves is always the same, the duration of combustion and the slope of the main curve may change with engine conditions. Curves *a* and *b*, Fig. 8-18, show the influence of the point of ignition of the charge; curve *c* shows the retarding influence of increasing the air-fuel ratio; and curve *d* shows the influence of the density of the charge.

8-5. Combustion Chambers. *Requirements.*—The basic requirements of a good combustion chamber together with the accompanying desirable characteristics are:

1. *High power output,* which requires:
   (*a*) a high compression ratio;
   (*b*) small or no excess air;
   (*c*) a complete utilization of the air—no dead pockets;
   (*d*) good turbulence;
   (*e*) room for large inlet and exhaust valves to obtain a high volumetric efficiency;
   (*f*) streamlining in order to reduce the pressure drop and to increase further the volumetric efficiency.
2. *High thermal efficiency,* which in turn requires:
   (*a*) a high compression ratio;
   (*b*) a small heat loss during combustion, which means
   (*c*) a small surface-volume ratio, a compact shape.
3. *Smooth engine operation,* which requires:
   (*a*) a moderate rate of pressure rise during combustion;
   (*b*) absence of detonation, which in turn means
   (*c*) compact combustion space, short distances for flame travel;
   (*d*) proper location of the spark plug;
   (*e*) good cooling of the charge during the compression stroke, a low temperature of the charge near the ignition point.

All the above enumerated characteristics are either self-evident or previously discussed. The characteristics 2*b* and 3*e*, which seem to contradict one another, refer to two different phenomena: a small relative surface during combustion and walls with a high coefficient of heat transfer during the entire compression.

Types of Combustion Chambers.—Figure 8-19*a*[1] shows a so-called T-head chamber, used extensively in older multicylinder engines. It fulfills requirement 1*d* and 3*e* and gives a short cylinder block but is less satisfactory in nearly all other respects, particularly in respect to detonation.

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[1] TAYLOR and TAYLOR, *op. cit.*, p. 86.

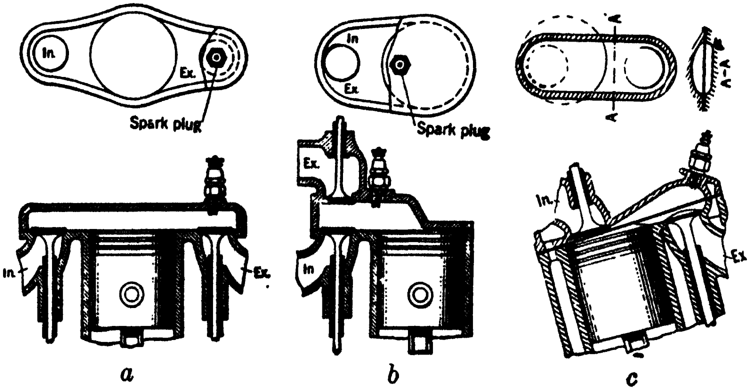

FIG. 8-19.—T- and F-head combustion chambers.

The early F-head arrangement, Fig. 8-19*b*,[1] is an improvement over the T-head one in several respects but presents difficulties in the design of the valve-operating mechanism.

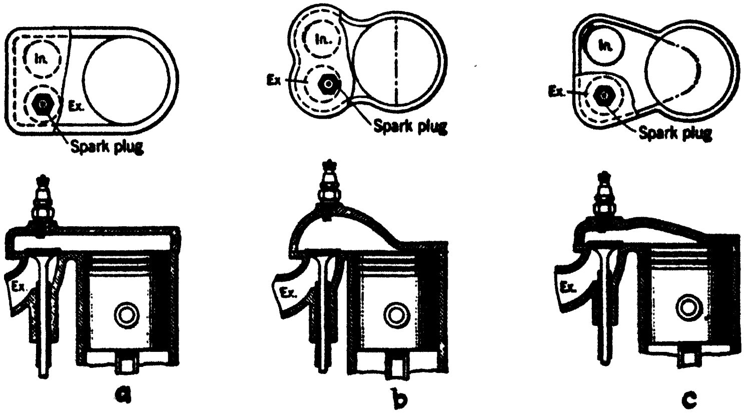

FIG. 8-20.—L-head combustion chambers.

A streamlined offset F-head chamber, Fig. 8-19*c*;[2] has a long flame travel owing to the special spark-plug location to give smoothness, and a relatively low rate of pressure rise, without an objectionable detonation.

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[1] TAYLOR and TAYLOR, *op. cit.*, p. 86.

[2] WHATMOUGH, W. A., Combustion-chamber Design in Theory and Practice, *SAE Journal*, vol. 25, p. 249, 1929.

The development of the L-head arrangement, Fig. 8-20*a*,[1] was an improvement over the T-head engine in respect to the valve operating mechanism and, to a certain extent, in detonating tendency. The arrangement of Fig. 8-20*b*[1] was a further step, giving a high turbulence and high power output. Figure 8-20*c*[1] shows a streamlined arrangement with a good resistance to detonation due to a rapid heat dissipation from the last portion of the charge to burn. The high turbulence accounts for a high power output equal to or better than the typical valve-in-head arrangements, Fig. 8-21*a*[1] and *b*, which have an inherently high turbulence and were formerly considered the best for high-power output.

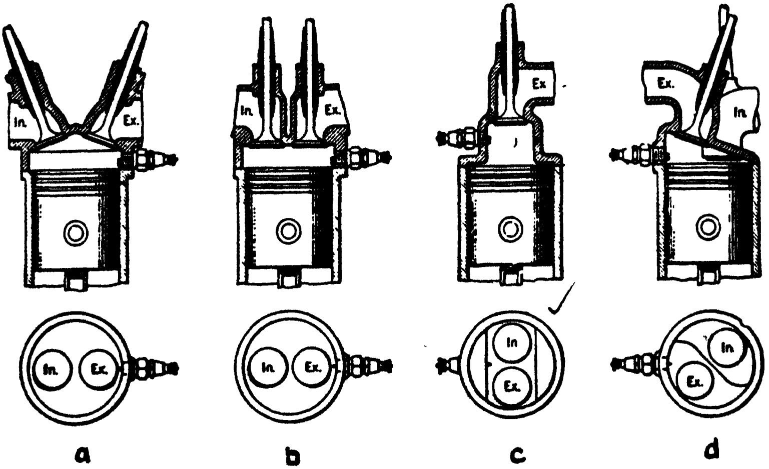

FIG. 8-21.—Valve-in-head combustion chambers.

However, the late valve-in-head engines, Figs. 8-21*c* and 8-21*d*, have a still better turbulence and volume control. The spark plug in both engines is located to promote a rapid burning of the first portion of the charge. The roof area of Fig. 8-21*d*, which is smaller than that on Fig. 8-20*c*, gives a smaller heat loss for the valve-in-head engines.

Finally, Fig. 8-22 shows a combustion chamber suitable for a horizontal natural-gas engine, in which the question of turbulence is important and the compression ratio is limited not by detonation but by preignition.

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[1] TAYLOR and TAYLOR, *op. cit.*, p. 86.

Material.—High-compression cylinder heads are made of materials with a high heat-transfer coefficient, such as aluminum alloys, in order to obtain a relatively cool combustion-chamber wall and thus lower the knocking tendency. However, improper metal distribution or poor circulation of the cooling liquid may cause the formation of hot spots and thus destroy the advantages of the aluminum head.

Surface Condition.—Combustion chambers with polished surfaces have a greater knocking tendency than those with surfaces coated with a light carbon deposit. This is due probably to the difference in heat-absorbing characteristics. A deposit of hard carbon results in a hotter surface and in an increase of the compression ratio, both of which increase the tendency to knock.

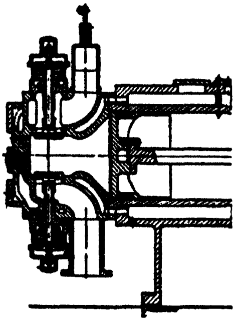

FIG. 8-22.—Combustion chamber of a 17-in. × 18-in. gas engine.

8-6. Problems.—1. Using formula (8-1) find the ignition temperatures of a theoretical air-gasoline mixture at 60 and 75 psig.

2. Using formula (8-1) find the ignition temperatures of an air-gasoline mixture, $r_{af} = 15.2$, compressed in an engine to 70 and 90 psig when the initial conditions were $p_1 = 1$ psi vacuum and $t_1 = 175$ F.

3. Using formula (8-1) find the ignition temperatures of an air-benzene mixture, $r_{af} = 15.3$; the initial pressure is 14.0 psia; the initial temperature is $t_1 = 180$ F; the compression ratios are (*a*) 6.25:1 and (*b*) 6.75:1.

4. Determine whether there is danger of preignition in the two cases of problem 2.

5. Determine the difference between the ignition temperatures in problem 3 and the compression temperature when the crank is 30° before top dead center.

6. Determine the theoretical angles of optimum spark advance for a 2¾-in. × 3½-in. engine running at 1200, 2500, and 3700 rpm.

7. Determine the duration of combustion in seconds for the engine in problem 6 if combustion is completed in (*a*) 12° of crank travel after the dead center at 1200 rpm, (*b*) 16° at 2500 rpm, and (*c*) 18° at 3700 rpm.

8. Determine the theoretical angles of optimum spark advance for a 3¾-in. × 4½-in. engine running at 1800, 2200, and 2600 rpm.

9. Determine the duration of combustion in seconds for the engine in problem 8 if combustion is completed: (*a*) 15° of crank travel after dead center at 1800 rpm, (*b*) 15.5° at 2200 rpm, and (*c*) 16° at 2600 rpm.

10. Compute the average ideal efficiency for an engine with a combustion time of 42° of crank travel with the ignition at 22° before dead center, a compression ratio 5.0, theoretical amount of air, $L/R = 4.5$, mass-burned curve, Fig. 8-16.

11. Compute the average ideal efficiency for the engine of problem 10 if the ignition is advanced to 32° before dead center.

12. Compute the average ideal efficiency for the engine of problem 10 if the ignition is advanced to 40° before dead center.

13. Compute the average ideal efficiency for an engine with a combustion duration of 49° of crank travel, with the ignition set at 25° before dead center; the compression ratio is $r = 5.5$; 5 per cent excess air is present; the connecting rod is 2.4 times the piston travel.

14. Compute the average ideal efficiency for the engine of problem 13 if the spark is advanced to 35° before dead center.

15. Construct rate-of-burning curves for the mass-burned curves *a* and *b* of Fig. 8-18.

16. Construct rate-of-burning curves for the mass-burned curves *a* and *c* of Fig. 8-18.

17. Construct rate-of-burning curves for the mass-burned curves *a* and *d* of Fig. 8-18.

18. Compare the combustion chamber of Fig. 8-19*b* in respect to compliance with each of the requirements for a good combustion chamber as enumerated in Sec. 8-5.

19. Compare the combustion chamber of Fig. 8-20*b* in respect to compliance with each of the requirements for a good combustion chamber as enumerated in Sec. 8-5.

20. Compare the combustion chambers Fig. 8-20*c* and 8-21*d* as to which one should give higher output, higher thermal efficiency, and smoother burning; state the reasons.

CHAPTER 9

HANDLING OF FUEL OILS

9-1. Fuel-oil Injection. *Methods.*—There exist two basic methods of fuel injection in compression-ignition oil engines—*air injection* and *airless injection.* The latter method is known under different names, such as *mechanical, solid,* or *hydraulic* injection.

Airless-injection methods can be subdivided into four groups: (*a*) constant pressure, or common-rail system, (*b*) jerk-pump system, (*c*) distributor system, and (*d*) individual-pump system in connection with a precombustion chamber.

The first two systems use high injection pressures to atomize the fuel; in the last two systems, some designs use high and some low injection pressures. In addition to the enumerated main types there exist a number of intermediate types having some characteristic of one and some of another system.

Air injection was first successfully developed by Rudolf Diesel. It requires a compression pressure of about 500 to 525 psi and air injection pressure of about 1000 psi. Its advantages are very good atomization and distribution of the fuel, resulting in comparatively high mean effective pressures; its disadvantages are complication of the engine by a multistage air compressor, which often is a source of trouble and in any event requires expert attention and at the same time lowers the mechanical efficiency of the engine. The latest achievements in airless-injection engines seem to spell doom for the air-injection engines, and the number of manufacturers of this type decreases with every year.

Constant-pressure injection, also called *Vickers,* or *common-rail,* or *accumulator system,* was developed by the Vickers Company in England and was the first satisfactory mechanical-injection system. It consists of a system of pipes, which connect the high-pressure oil pump or pumps with the fuel or spray nozzle in each cylinder. A sufficient accumulator action maintaining a more or less constant pressure in the system is obtained by the compressibility of the fuel in the pipes and other parts such as the fuel strainer and pressure-relief-valve body. The fuel is admitted into the cylinders by lifting the fuel valve, called also *fuel needle* on account of its shape, by a cam mechanism driven from the engine crankshaft, Fig. 9-1.

The compression pressure of the air charge is lower than in air-injection Diesel engines, usually between 370 and 420 psi. The ignition pressure goes up to 550 to 600 psi, partially because it is difficult to obtain a really constant-pressure combustion with the mechanical injection of the fuel, partially because the increase of the maximum pressure gives an increase of the thermal efficiency. Fuel-injection pressures range from 3200 to 5000 psi and in some engines are as low as 1700 psi.

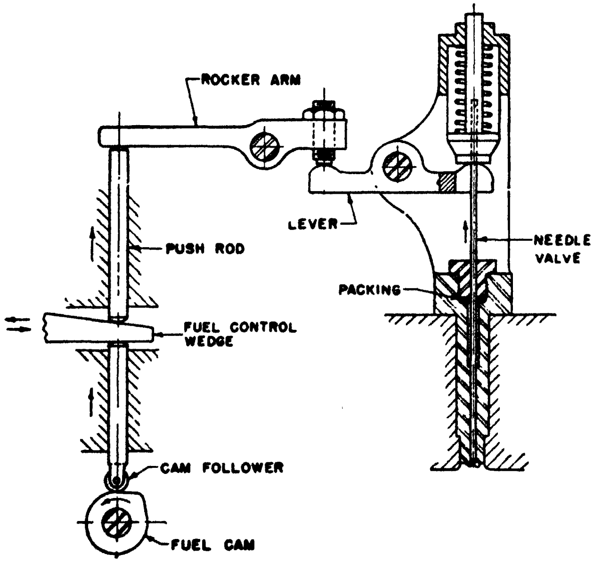

FIG. 9-1.—Control of fuel admission in a common-rail injection system.

Jerk-pump injection differs from the constant-pressure injection in both design and operation of the pumps and fuel injectors. Each injector has a separate pump, and the injector becomes a spring-loaded, hydraulically operated automatic valve; no mechanism is required to operate it. The system lends itself equally well for all cylinder sizes and engine speeds. However, for very small engines it is rather expensive.

Distributor System.—In this system the fuel oil is supplied, through rotating distributors, to individual metering pumps and from there to the engine cylinders. Of the several distributor systems encountered in American engines, that of the Cummins engines is probably the best known one.

Precombustion Chambers.—The compression space is subdivided into two parts, one formed by the space between the piston top and the cylinder head and the other formed by a special recess in the cylinder head and connected with the main chamber by a more or less restricted passage. The fuel is injected into the precombustion chamber sometimes during the compression stroke, the earlier the smaller the passage to the precombustion chamber. Thus the fuel is given time to evaporate before it ignites in the precombustion chamber. On account of the insufficient amount of air, only partial combustion occurs, but it is sufficient to raise the pressure and to throw out the rest of the fuel into the main chamber, where combustion is completed.

The oil-injection pressure may be considerably lower, 1000 to 1500 psi. However, some engines use high-pressure injection.

The advantages of this system are: the timing of injection does not need to be so exact; the fuel charge is given a longer time element to become ready for combustion, and efficient turbulence is created by the precombustion discharge. All these features are very helpful in variable- and high-speed engines as used in automobiles, trucks, and tractors. The disadvantages are: not as good control of the combustion and difficulty in obtaining high mean effective pressures.

Hot-surface-ignition engines have an injection system similar to that of precombustion-chamber engines but use still lower injection pressures.

Governing.—In most oil engines the amount of fuel injected is regulated by the pump, whose action is under the control of the governor.

In mechanical-injection constant-pressure engines the governor acts upon the fuel-valve lift mechanism by changing the duration and height of the lift of the fuel needle by means of the fuel-control wedge, Fig. 9-1, which increases or decreases the lost motion of the push rod.

Thus the problem of governor control is really a problem of metering the fuel.

Requirements.—The main requirements which a fuel-injection system must fulfill are:

1. Accurate metering of small amounts of fuel oil.
2. Proper timing of the fuel injection.
3. Control of the rate of fuel injection.
4. Atomization of the fuel in accordance with the type of the combustion chamber used.
5. Good distribution of the fuel in the combustion space.

Fuel per Cycle.—The amount of oil injected per cycle depends upon the piston displacement and load factor. To obtain a complete combustion at full load the air supply must be about 50 per cent greater than the theoretical amount and sometimes even more. In engines with high-turbulence combustion chambers, the excess air may be reduced down to about 30 per cent. If more fuel is injected than can be burned completely, the specific fuel consumption and cost of power goes up. If less fuel is injected, the mean effective pressure decreases and a smaller power will be developed per cubic foot of displacement, again increasing the cost of power. The proper amount of fuel giving the lowest cost of power for a given engine must be determined by testing.

If W_h is the fuel consumption in pounds per horsepower-hour, N is the number of horsepowers per combustion space, i the number of strokes per cycle, and n designates revolutions per minute, then the weight of fuel per cycle

$$W_c = W_h N i/120n \tag{9-1}$$

and the volume in cubic inches is

$$V_c = 1728W_c/62.4\gamma = 27.7W_c/\gamma \tag{9-2}$$

where γ is the specific gravity of the fuel oil, referred to water.

Example 9-1.—Determine the quantity of fuel to be injected per cylinder per cycle in a six-cylinder four-stroke-cycle air-injection engine developing 720 hp at 180 rpm with a fuel rate of 0.40 lb/hp-hr.

From expression (9-1)

$$W_c = (0.40 \times 720 \times 4)/(120 \times 180 \times 6) = 0.00889 \text{ lb/cycle}$$

If oil of 24° API is used, its specific gravity γ, by equations (3-2),

$$\gamma = 141.5/(131.5 + 24) = 0.911$$

and by equation (9-2), at atmospheric pressure,

$$V_c = 27.7 \times 0.00889/0.911 = 0.27 \text{ cu in./cycle}$$

Compressibility of Oil.—The coefficient of compressibility is

$$b = \frac{V_1 - V_2}{V_1(p_2 - p_1)} \tag{9-3}$$

where b is approximately 3×10^{-6} and p_1 and p_2 are expressed in psi.

Volumetric efficiency of a pump depends upon:

(*a*) clearance between the pump plunger and barrel
(*b*) injection pressure,
(*c*) number of strokes per minute,
(*d*) viscosity of the fuel,
(*e*) pressure of the fuel supplied to the pump.

Figure 9-2 shows that the most important factor is the clearance;[1] then come the injection pressure and speed. The influence of the viscosity is not very important, with present precision-manufactured pumps with clearances not greater than 0.0001 in. An increase of the fuel supply pressure from 1.5 psig to 45 psi increases the volumetric efficiency from 1 to 2 per cent for a viscosity of 60 SSU and over and slightly more for fuels of 35 to 60 SSU. In general, with low injection pressures, 1500 to

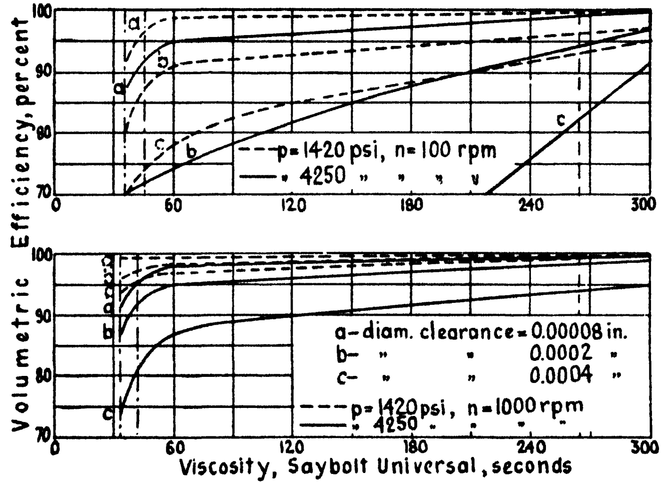

FIG. 9-2.—Influence of various factors on the volumetric efficiency of a high-pressure fuel-injection pump.

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[1] BAIERLEIN, G. W., Effect of Viscosity of Fuel Oil on Injection Equipment, *Diesel Power and Transportation*, vol. 20, No. 2, p. 136, February, 1942.

1700 psi, and a normal clearance, the volumetric efficiency is close to 95 per cent and drops with the injection pressure down to about 85 per cent at 4250 psi. With a very low fuel viscosity, the volumetric efficiency may drop down to 75 per cent.

9-2. Air Injection.—*Metering* of the fuel—controlling the quantity of the fuel delivered per cycle in accordance with the engine load—is done by the fuel-delivery pump, using one of the following methods:

(*a*) changing length of the pump stroke,

(*b*) by-passing fuel at the beginning of the delivery stroke, or

(*c*) by-passing fuel at the end of the delivery stroke.

If x designates the useful fraction of the pump stroke s, and d its diameter, then the useful pump displacement can be presented as

$$0.7854 d^2 s \eta_v x = V_c + (V_1 - V_2) \qquad (9\text{-}4)$$

where V_1 is the volume of the oil in the pump system at the suction pressure and V_2 the volume of the oil at the pressure of delivery. The volumetric efficiency η_v in a new pump is about 0.96 and in time may go down to 0.90.

Example 9-2.—Determine the size of the pump for the engine of Example 9-1. The volume of the oil in the suction chamber is 0.9 cu in.; the discharge pressure is 1100 psig; the volume of the oil in the injector is 0.275 cu in.

$$V_1 = 0.9 + 0.27 = 1.17 \text{ cu in.}$$

By expression (9-3) the volume decrease due to compressibility is,

$$V_1 - V_2 = bV_1(p_2 - p_1) = 3 \times 10^{-6} \times 1.17 \times 1100 = 0.004 \text{ cu in.}$$

Assuming in accordance with average practice that $s = 2d$, $\eta_v = 0.94$, and at full load in this pump type $x = 0.5$, from equation (9-4),

$$d = \sqrt[3]{(0.27+0.004)/(0.7854\times2\times0.94\times0.5)} = \sqrt[3]{0.371} = 0.718 \text{ or } \tfrac{23}{32} \text{ in.}$$

the total stroke will be $\frac{23}{32} \times 2 = 1\frac{7}{16}$ in.

Fuel Pumps.—Their function is to deliver a predetermined quantity of fuel oil into the nozzle against a pressure of the injection air, some time during the exhaust, suction, or compression stroke. These pumps are of the plunger type. Formerly the plungers were sealed by packing glands, but in newer designs they are of the packingless type, with the highly polished plunger very closely fitted in the cylinder. The plunger is operated either by a cam and returned by a spring or by an eccentric. The pump in Fig. 9-3 is used on a large oil engine, and the metering is effected through by-passing of a part of the fuel during the beginning of the delivery stroke. The eccentric *e* moves the plunger *p* through the crosshead *c*; *s* is the suction and *d* the delivery valve; *f* is the fuel-line connection; *a* is an auxiliary plunger which is connected by links *g*, *h*, *i*, *k* to the crosshead *c*; plunger *a* in its upper position prevents the closing of the suction valve *s* when the main plunger *p* starts on the downward stroke, and the fuel delivery begins only after plunger *a* has moved down far enough to allow the suction valve *s* to close. The governor, by turning an eccentric *b*, lowers or raises the link *g* and thus changes the moment when the plunger *a* releases valve *s*.

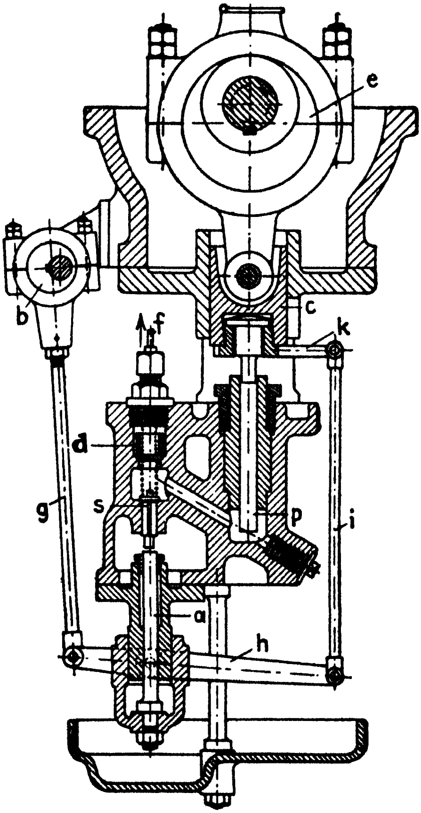

FIG. 9-3.—Fuel pump for an air-injection engine.

Rate of Injection.—Some information in respect to the rate of injection can be obtained from Fig. 9-4.[1] The abscissas represent crank angles φ proportional to time τ; the ordinates give the percentage of oil injected and burned. Curve $y = f_1(\varphi)$ gives the amount of fuel injected, and curve $x = f_2(\varphi)$ the amount of fuel burned. The injection period extends over 40°, the combustion duration over 78°. The horizontal distances between the two curves give the time lag, which increases toward the end of combustion. The curve $dx/d\varphi = dy/d\tau$ gives the combustion velocity called *rate of burning*, and shows a maximum, as should be expected. The engine speed for which Fig. 9-4 was constructed was 216 rpm. This gives for the last burned fuel particle a lag of 0.029 sec. Under different conditions the x-curve may show a smaller lag at the beginning and a greater lag toward the end, curve x_2.

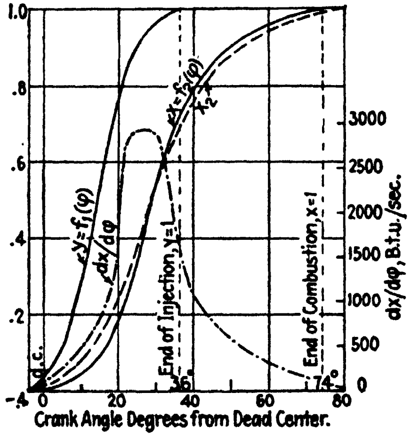

FIG. 9-4.—Combustion lag in oil engines.

Atomization.—The duration of combustion in an oil engine should last about 40° of crank travel and, taking into consideration unavoidable afterburning, not over 80°, which means 0.067 sec at 200 rpm and only 0.0074 sec at 1800 rpm. This exceedingly short time element requires a very fine atomization.

The theoretical velocity v of air flowing through an opening from a space with a pressure p_1 into a space with a pressure p_2 can be computed from the expression

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[1] NEUMANN, Untersuchungen an der Dieselmaschine, *Forschungsarbeiten*, Heft 245.

$$v = \varphi \sqrt{2g \frac{k}{k-1} RT_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]} \qquad (9\text{-}5)$$

or, substituting $g = 32.2$, $k = 1.4$, and $R = 53.34$,

$$v = 109.7\varphi \sqrt{T_1[1 - (p_2/p_1)^{0.286}]} \qquad (9\text{-}6)$$

where, if p_2 is smaller than the critical pressure $p_{cr} = 0.53p_1$, p_{cr} must be used instead of p_2 and φ is the velocity coefficient which varies with the resistance in the nozzle. To obtain proper atomization the nozzles have special appliances, which present a great resistance to the flow of air and oil and give for φ a value of 0.6 and down to 0.5. The amount of air necessary to inject and atomize the fuel is about 6 to 8 per cent of the fresh-air charge.

Comparing the energy applied to breaking up of the fuel into minute globules with the total energy of the air blast shows that only about 3 per cent of the energy is used for atomization proper. However, the rest of the energy is not entirely wasted as it creates vigorous turbulences which materially assist the combustion.

Fuel Distribution.—For complete combustion the fuel particles must be distributed uniformly through the whole combustion space. As this is actually impossible, excess air must be present in the cylinder to prevent incomplete combustion.

In air-injection engines the distribution of the fuel is accomplished through a thorough mixing of the injection air carrying the atomized fuel with the air in the combustion space.

Timing.—As shown on Fig. 9-4, in spite of a very fine atomization there is a lag between the time of injection and ignition; and the injection must start slightly before the dead center, some 2 to 10° of the crank travel, depending on the fuel characteristics and the speed, compression ratio, and size of the engine.

Fuel-injection valves, called *fuel nozzles*, can be divided in two main groups: the closed and the open type. The closed nozzles have a spring-loaded valve near the discharge orifice, and the fuel injection is controlled mechanically by means of cams, eccentrics, and levers.

With the open-type nozzle, the fuel is delivered into a space that is constantly in communication with the combustion chamber. From there the fuel is thrown out into the cylinder by a blast of air admitted through a mechanically operated valve, similar to that of a closed-type nozzle.

The main advantage of open nozzles is that the pump which delivers the fuel does not have to work against the high injection-air pressure, but only against the comparatively low pressure in the cylinder during the early part of the compression stroke. The serious disadvantage of the open-type nozzle is that they can be used only when located horizontally; otherwise they may cause dribbling.

Figure 9-5 shows a closed-type air-injection fuel valve known as the *Diesel* or *Augsburg* type. The fuel is delivered from the pump through a drilled hole *f* just above the atomizer; the latter consists of disks *d* with offset small holes and a cone *c* with grooves on the outside surface; the ring space *r* communicates with the high-pressure air bottle through the hole *a* shown in dotted lines; *p* is the flame plate made of heat-resistant steel. The needle valve *n* is lifted by the lever *l* and returned at the end of injection by a heavy spring *s*. The spring *z* and tube *t* keep the atomizer parts in place; *b* is a stuffing box with a screwed gland *g*.

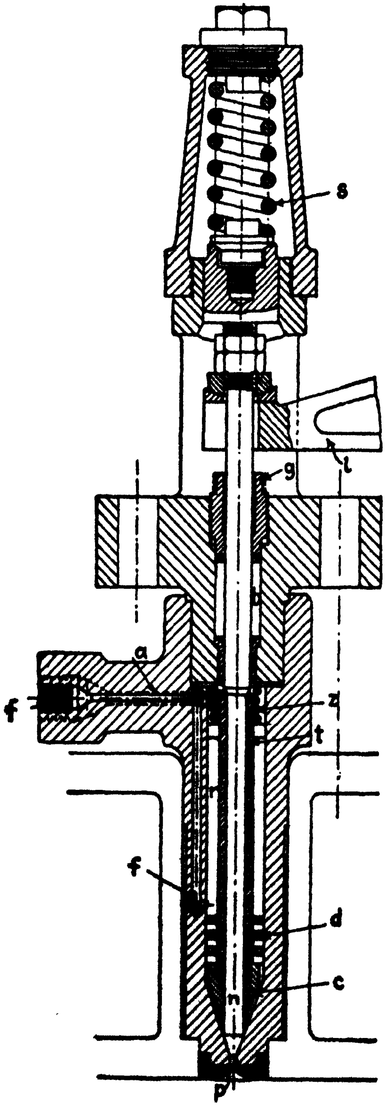

Fig. 9-5.—Closed air-injection fuel nozzle.

9-3. Airless Injection.—*Metering* of the fuel is done along the same lines as in air-injection engines; and the same methods of fuel-delivery control are used, except for constant-pressure mechanical-injection engines, in which the fuel is metered directly by the duration of opening of the fuel-injection valve itself.

Timing.—A peculiarity of airless injection is the time lag of ignition. Regardless of the shape and construction of the spray nozzle and of the height of the fuel pressure, airless injection generally gives a coarser atomization than air injection and requires a considerably earlier injection timing—from 10 to 30° before dead center. The proper angle of advance of the injection depends upon many factors, such as the characteristics of the fuel oil, method and details of the injection system, engine speed, and design of the combustion space, and can be determined only by actual testing.

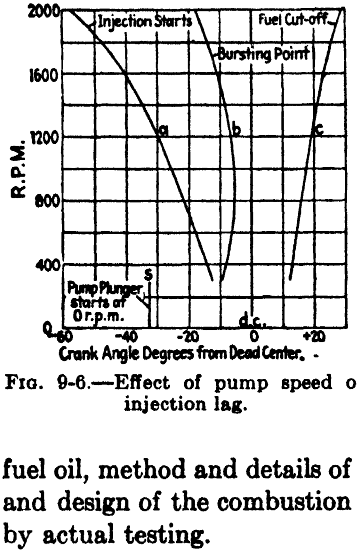

FIG. 9-6.—Effect of pump speed on injection lag.

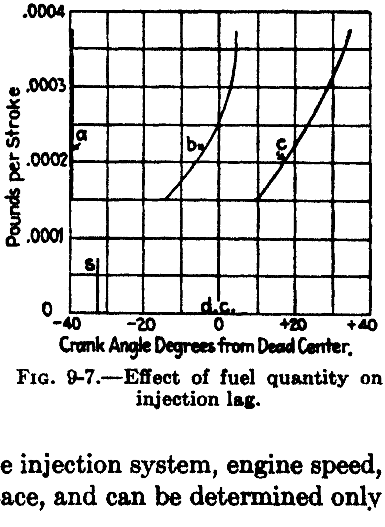

FIG. 9-7.—Effect of fuel quantity on injection lag.

Figure 9-6 gives an interesting picture of the influence of engine speed upon the lag caused by the injection mechanism. The tests were performed with a constant quantity of fuel injected, 0.00025 lb per stroke, with a cam-operated injection pump and a spring-loaded automatic injection valve, with an orifice $d = 0.025$ in. and valve-operating pressure 6000 psi. At 300 rpm, injection started 12° before dead center and was completed 12° after dead center. With an increase of the speed, the opening of the valve, as indicated by curve *a*, became earlier and the duration of injection, distance to curve *c*, which shows closing of valve, naturally increased but less than in a straight proportion to the speed, showing that the velocity of the oil flow increased with pump speed. Curve *b* shows the moment when the heavy spray burst and became finely atomized.

Figure 9-7 shows the results of tests with a constant speed of 1600 rpm and a variable weight of fuel injected per stroke. All curves are designated by the same letters as in Fig. 9-6. The injection advance remained constant at about 7°; the injection period increased with an increasing fuel amount per stroke and atomization of the fuel; the bursting point began later.

The angle of injection advance, in open-combustion-space mechanical-injection engines varies from 10 to 30°, and in precombustion-chamber engines it must be slightly earlier, 30 to 45° before dead center.

In most injection systems the angle of advance remains constant; however, some pumps are built so that the angle of advance is gradually increased with an increase of the load, and some, for use with variable-speed engines, have also provisions for an automatic or manual increase of the angle of advance with an increase of the engine speed.

Rate of Injection.—The rate of injection could be determined from a pressure-crank-travel diagram of the combustion process if the time lag of combustion were known.

Testing of nozzles in a special apparatus and measuring the actual rate of injection have shown that the same fuel pump, driven by the same cam, will give different rates of delivery with different types of nozzles. Typical rate-of-injection curves *a* and *b*, obtained with a Bosch pump and two different types of nozzles, are shown on Fig. 9-8. The curves of total fuel injected *a′* and *b′* are in general similar to the *y*-curve of Fig. 9-3.

The desirable rate of injection depends upon the air turbulence in the combustion chamber and the engine speed. In general, in order to prevent a high pressure rise in the engine cylinder, the rate of injection should be low at the beginning and increased afterward when the engine piston is approaching the dead center. Such a procedure is actually desirable at low rotative speeds and permits control of the maximum pressure in the cylinder. However, at high rotative speeds, in order to avoid an excessively early timing and low thermal efficiency, if burning extends considerably past dead center, high rates must be used from the beginning of the injection, and high combustion pressures are unavoidable.

Atomization in airless injection is produced by friction between the stream of fuel and the air in the combustion space. The liquid stream is ejected with a high velocity, created by the pressure of injection, through one or several small holes.

The mean diameter of the droplets into which the spray is broken up is approximately 0.001 in., decreasing with an increase of the injection pressure and also with a decrease of the nozzle-hole diameter.[1]

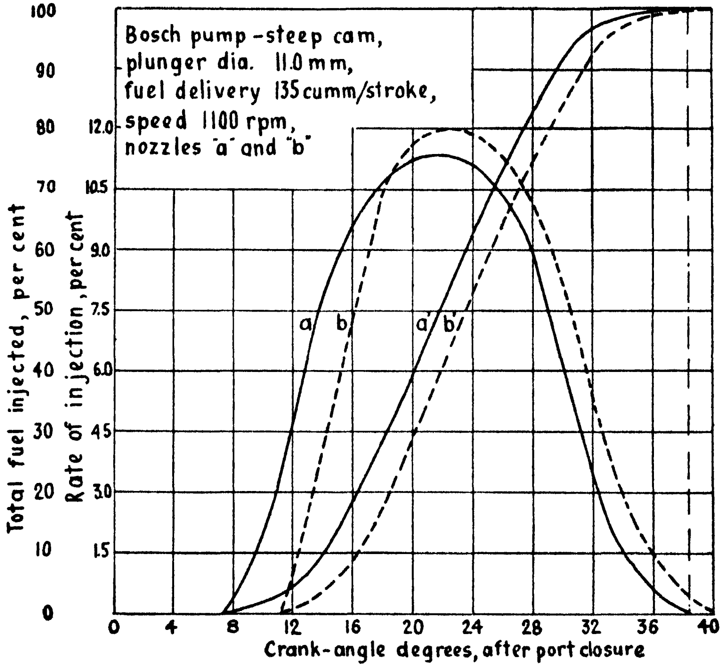

FIG. 9-8.—Rate-of-injection curves with pintle-type injection nozzles.

The velocity which produces proper atomization is, approximately, from 350 to 700 fps. It can be computed from the fundamental equation (7-1). Expressing the head h as a function of the pressure p_1, psi, of the fuel and p_2 of the space into which the fuel is injected and using for the weight of 1 cu ft of fuel the expression 62.48γ, where γ is the specific gravity, give

$$h = 144(p_1 - p_2)/62.48\gamma \tag{9-7}$$

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[1] LEE, D. W., Fuel Spray Formation, *Trans. ASME*, vol. 54, No. 20, p. 65, October, 1932.

Substituting expression (9-6) and $g = 32.2$ in equation (7-1) and reducing the numerical constants give for the velocity in feet per second

$$v = 12.18\varphi \sqrt{(p_1 - p_2)/\gamma} \tag{9-8}$$

where the velocity coefficient φ can be determined best from the discharge coefficient μ and the contraction coefficient α, $\varphi = \mu/\alpha$. The contraction coefficient for the type of holes used in nozzles may vary from 0.90, when the edges are sharp, up to 0.95, when they are slightly chamfered.

For oil of average viscosity at 80 F, the coefficient of discharge μ through small orifices depends upon the ratio p_1/p_2. It is about 0.75 for $p_1 = 10p_2$, 0.81 for $p_1 = 5p_2$, and 0.87 for $p_1 = 3.5p_2$ and down to $1.4p_2$. For $p_1 > 1000$ psi, the pressure and temperature of the oil practically do not affect μ. For orifices with a diameter above 0.02 in., the coefficient μ decreases with an increase of the diameter.[1]

Example 9-3.—Determine the velocity of injection in an airless fuel nozzle with one hole, when the oil pressure is 1700 psi and the pressure in the combustion chamber is 440 psi; the oil is 32° API.

$$p_1/p_2 = 1714.7/454.7 = 3.77$$

therefore

$$\mu = 0.87 - (0.87 - 0.81)(3.77 - 3.5)/(5.0 - 3.5) = 0.859$$

and assuming that $\alpha = 0.95$, gives

$$\varphi = 0.859/0.95 = 0.905$$

The specific gravity, from expression (3-5),

$$\gamma = 141.5/(131.5 + 32) = 0.866$$

Now, by equation (9-8),

$$v = 12.18 \times 0.905 \sqrt{(1700 - 440)/0.866} = 420 \text{ fps}$$

The size of the orifice through which the fuel is injected can be determined from the equation

$$W_c = \alpha A v t w \tag{9-9}$$

where A is in square feet, v is velocity, fps, t is duration of injection, sec, and w is specific weight, lb/cu ft. Expressing the area of the orifice by a in square inches, substituting for the velocity expression (9-7) changed to inch per second, substituting for $w = 62.4\gamma$ and $\mu = \alpha\varphi$, and solving for a give

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[1] JOACHIM, W. F., *NACA Rept.* 224, 1925.

$$a = 0.1895 W_c/\mu t \sqrt{\gamma(p_1 - p_2)} \tag{9-10}$$

Example 9-4.—Determine the diameter of the fuel orifice for a four-stroke engine developing 15 hp/cylinder at 1800 rpm using 0.44 lb/hp-hr of 32° API oil; the desired duration of the injection is 32° of crank travel; other data as given in Example 9-3.

By equation (9-1)

$$W_c = 0.44 \times 15 \times 4/(120 \times 1800) = 0.000122 \text{ lb/cycle}$$

The duration of injection in seconds

$$t = (60/1800)(32/360) = 0.00296 \text{ sec}$$

The specific gravity from expression (3-5), as in Example 9-3, $\gamma = 0.866$ and by expression (9-10), with other data from Example 9-3,

$$a = 0.1895 \times 0.000122/[0.859 \times 0.00296 \sqrt{0.866(1700 - 440)}] = 0.000275 \text{ sq in.}$$
$$d = \sqrt{0.000275/0.7854} = 0.0187 \text{ or } 0.019 \text{ in.}$$

Fuel Distribution.—In airless injection, fuel distribution is accomplished either by a nozzle tip which gives a cone-shaped spray, or by a tip with several holes, five to eight, or in larger engines by the use of two or more separate fuel nozzles.

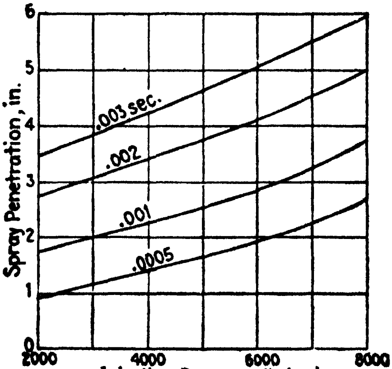

FIG. 9-9.—Effect of injection pressure on penetration.

To obtain complete combustion, as in air injection, a certain excess air is necessary.

Penetration.—A prerequisite of fuel distribution is proper penetration. Penetration is proportional to the energy imparted to the fuel particles by the pressure difference $p_1 - p_2$, the same as atomization, but it is influenced in the opposite way by the size of the hole—it decreases with a decrease of the hole size, whereas atomization is improved by this decrease. Thus the size of the hole must be determined as a compromise between proper atomization and sufficient penetration.

The influence of pressure p_1 and p_2 as found experimentally is shown in Fig. 9-9: gas pressure p_2 in the injection chamber was 200 psi; penetration increases with the pressure difference $p_1 - p_2$ and is retarded by the gas resistance.

Figure 9-10 shows in two different systems of coordinates the rapid decrease of penetration with an increase of the air pressure p_2 resulting in an increase of the air density.

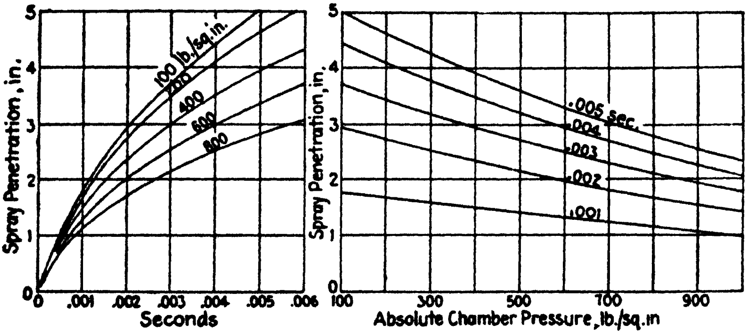

FIG. 9-10.—Effect of chamber pressure on penetration. Fuel pressure, 8000 psi; orifice diameter, 0.022 in.

Penetration, or rather the time element necessary for the travel of fuel through a certain distance, limits the size of the engine cylinder with an increase of the engine speed, as a smaller time element is then allowed for each event.

The influence of the type of nozzle is clearly shown in Fig. 9-11.[1] At the same time the diagram shows that the injection pressure increases penetration, but only slightly, and the effect of fuel viscosity is almost negligible.

Turbulence in the combustion space aids fuel distribution, lowers fuel consumption, and increases the engine power materially. The design of the intake ports, combustion space, and top of the piston may create additional turbulences which it is difficult to measure directly, but its influence can be established by measuring the power increase and fuel consumption.

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[1] BAIERLEIN, *op. cit.*, p. 143.

Tests[1] have demonstrated that additional turbulences can increase the mean indicated pressure up to 17 per cent and at the same time lower the fuel consumption by about 15 per cent.

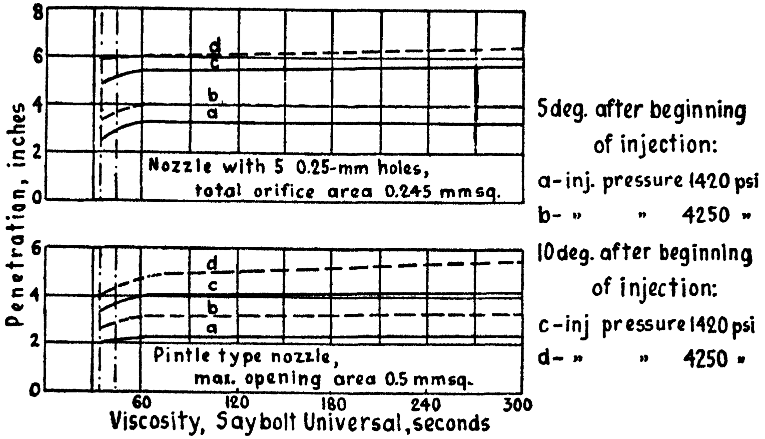

Fig. 9-11.—Influence of various factors on fuel penetration.

Work of Injection.—This work may be calculated using the *p-V* diagram for the injection process. The compression of the fuel may be assumed to be represented by a straight line on this diagram, Fig. 9-12. The fuel trapped in the pipe line, between the discharge valve of the pump and the injection orifice, will have a pressure somewhat lower than the compression pressure in the cylinder.

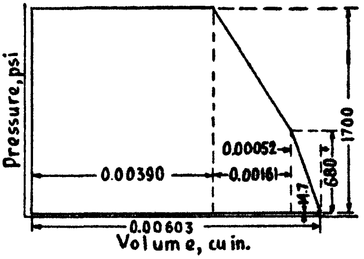

Fig. 9-12.—Pressure-volume diagram for the fuel-injection process.

Example 9-5.—Determine the work of injecting the fuel for the conditions of Examples 9-3 and 9-4. The fuel volume in the pump is 0.252 cu in.; the fuel line is of ⅛-in. bore and 20 in. long; the residual pressure in the line is about 60 per cent below the injection pressure.

The useful displacement is found by equation (9-2) as

$$V_e = 27.7 \times 0.000122/0.866 = 0.0039 \text{ cu in.}$$

The volume of the oil in the pipe line is

$$V_l = 0.7854 \times 0.125^2 \times 20 = 0.2450 \text{ cu in.}$$

The volume of the oil in the nozzle is 0.025 cu in.

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[1] Kemper, C., *NACA Tech. Note* 242, 1926.

The residual pressure is $p = 1700 \times 0.4 = 680$ psig or 694.7 psia. The volume change for a pressure increase of 680 psi is

$$V_1 - V_2 = (0.252 + 0.0039)680 \times 3.0 \times 10^{-6} = 0.00052 \text{ cu in.}$$

The volume change for a further increase in pressure to 1700 psig is

$$V_1' - V_2' = (0.2559 - 0.00052 + 0.245 + 0.025)(1700 - 680) \times 3.0 \times 10^{-6} = 0.00161 \text{ cu in.}$$

Thus the total pump displacement is

$$V_d = 0.0039 + 0.00052 + 0.00161 = 0.00603 \text{ cu in.}$$

The total work from the p-V diagram, Fig. 9-12,

$$W = [(0.00052/2 + 0.00551)680 + (0.00161/2 + 0.0039)(1700 - 680)]144/1728 = 0.727 \text{ ft-lb}$$

with 1800 rpm of the engine the injection work per minute

$$W_i = 0.727 \times (1800/2) = 654 \text{ ft-lb/min}$$

Since the work developed per cylinder is 15 hp, the relative injection work in per cent

$$W_{ir} = 654 \times 100/(15 \times 33{,}000) = 0.132 \text{ per cent}$$

The actual work required will be about two times larger because of leakage and friction losses of the pump mechanism. The work will also be larger with a higher injection pressure.

9-4. Fuel-injection Equipment. Injection Pumps.—The pumps are of the packingless plunger type, with each plunger individually lapped in with its barrel. In most pumps the forward, or discharge, stroke is produced by a cam, and the return, or suction, stroke by a spring. The Ex-Cell-O pump uses a wobble plate instead of a cam.

The basic methods of controlling the quantity of the fuel delivered are as follows:

1. Variable pump stroke.
2. By-passing a variable fuel quantity at the end of the pump stroke by
   (a) a helix groove on the plunger, or
   (b) a hydraulically operated by-pass valve, or
   (c) a mechanically operated by-pass.

Variable-stroke Pump.—A Deco variable-stroke pump for a four-cylinder engine is shown on Fig. 9-13. The downward movement of the plunger *p* under the action of the spring *e* creates a vacuum over the plunger, which opens the suction valve *s*, thus filling the barrel *b* with fuel. When *p* reaches its lowest position, the suction is relieved and valve *s* is closed by its spring. During the upward stroke of *p* under the action of the cam *c*, the fuel is at first compressed and then discharged through the valve *d* to the injector and through it into the combustion chamber. The amount of fuel injected depends upon the length of the plunger stroke.

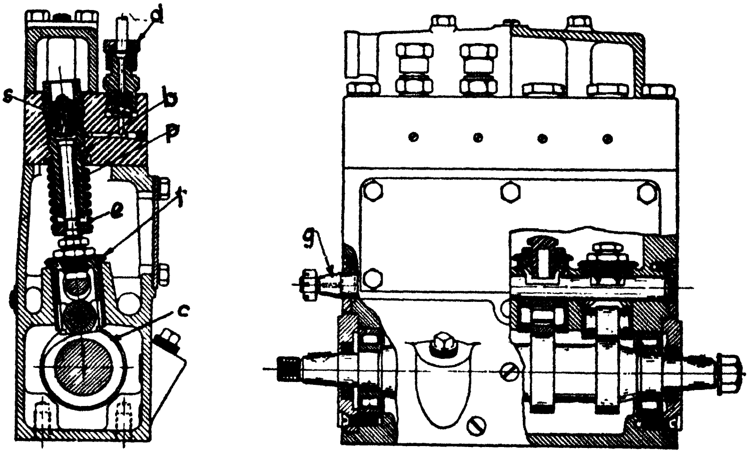


FIG. 9-13.—Four-cylinder Deco injection pump.

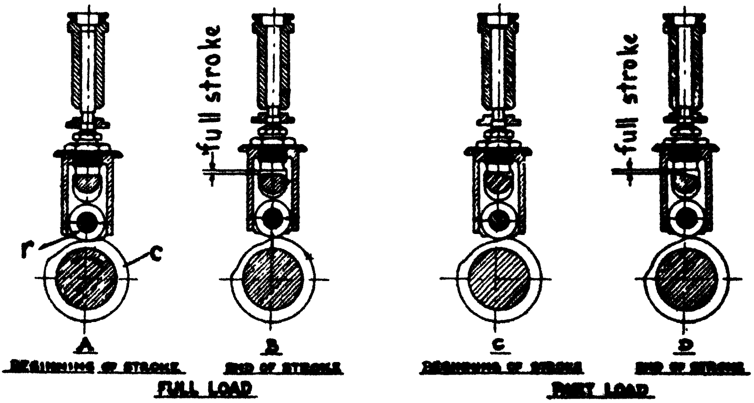


FIG. 9-14.—Operation of a variable-stroke pump.

The turning of the control shaft *g* causes its high edge to raise the tappet assembly *t* of each pump, thus decreasing the plunger motion downward and retarding the movement when the cam *c* comes in contact with the roller *r*, Fig. 9-14. By turning the shaft *g* to its limit position, the roller *r* may be raised so high that it will not be contacted by the cam *c* and injection will be stopped.

Helix By-pass Pumps.—This method has the advantages of being very simple mechanically and also of being the least affected by the viscosity of the fuel oil. It is used by several manufacturers.

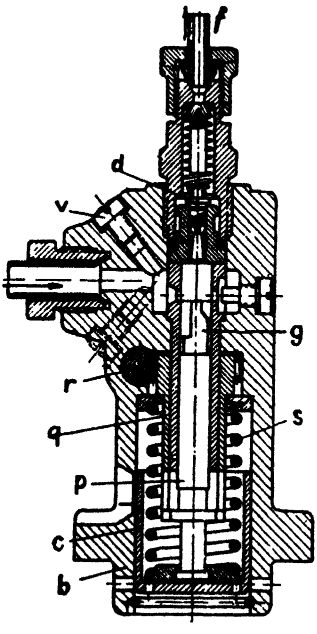


FIG. 9-15.—Bosch individual injection pump.

Figure 9-15 shows the upper part without the driving mechanism of a Bosch injection pump. This pump has no suction valve, and a spiral groove *g* on the packingless plunger *p* acts as a pressure-relief valve toward the end of the discharge stroke by connecting the space above the piston with the suction chamber. The governor moves a rack *r* engaged with gear teeth on the sleeve *q*, which is slidingly connected with the plunger *p*. The turning of the plunger *p* by the rack *r* changes the time of pressure release by the groove *g* and the amount of fuel delivery. The driving mechanism, a cam with a contour that can be selected according to the desired rate of injection, pushes the sleeve tappet *c* upward; and the spring *s* pushes *c* together with *p* downward.

The drive mechanism for such a pump is shown in Fig. 9-16. However, this particular mechanism has a provision to change, advance, or retard the beginning of the fuel injection, depending upon the ignition quality of the fuel used. Low-cetane fuels will knock with advanced injection timing, whereas high-cetane fuels require advanced timing for best performance and economy; hence this adjustment is called a *cetane selector*. The timing is changed by hand by shifting the fuel-cam roller, to the left for advancing or to the right for retarding, by turning the eccentrically supported fulcrum of the cetane selector.

Pumps with Mechanically Operated By-pass.—Figure 9-17 shows an Ex-Cell-O fuel-injection pump for a six-cylinder engine;

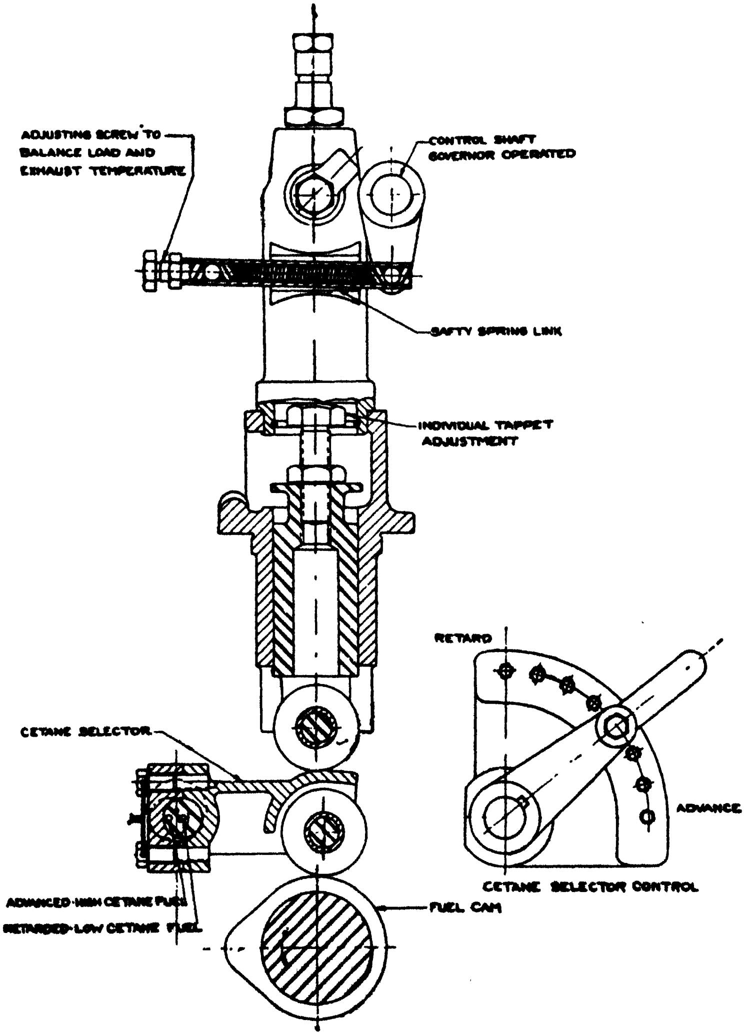

ADJUSTING SCREW TO BALANCE LOAD AND EXHAUST TEMPERATURE
CONTROL SHAFT GOVERNOR OPERATED
SAFTY SPRING LINK
INDIVIDUAL TAPPET ADJUSTMENT
RETARD
CETANE SELECTOR
ADVANCE
CETANE SELECTOR CONTROL
ADVANCED-HIGH CETANE FUEL
RETARDED-LOW CETANE FUEL
FUEL CAM

Fig. 9-16.—Pump drive with cetane selector. (*Courtesy of Clark Brothers Company.*)

the fuel enters through the passage Y and ports C into six pump cylinders B; the rotary valve W, with a recess and a raised trapezoidal land X, acts as a distributor. The delivery stroke of each of the plungers *D* is obtained from the rotating wobble plate *R* through push rods *J*. The return stroke is effected by a compression spring. The drive of the whole pump is through the drive gear *M*. When the plunger *D* begins to move to the left, port *C* is open, Fig. 9-18*a*, and no fuel is injected; when, because of the rotation of *W*, the land *X* begins to close the port, Fig. 9-18*b*, injection begins and lasts while *X* keeps the port *C* closed; when the land *X* begins to uncover the port, Fig. 9-18*c*, the port is opened, the fuel pressure drops and fuel injection ceases. When the governor weight *P*, acting through the rod *O*, moves valve *W*, with the land *X* axially to the left, the duration of injection and hence the amount of fuel injected

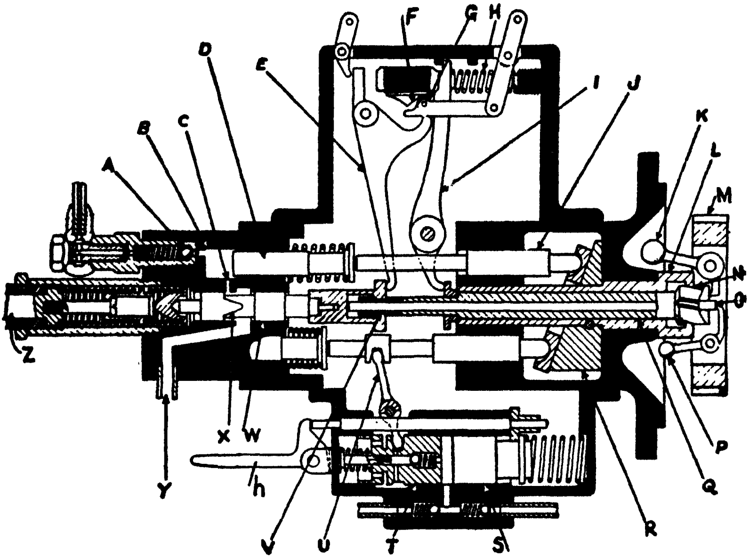
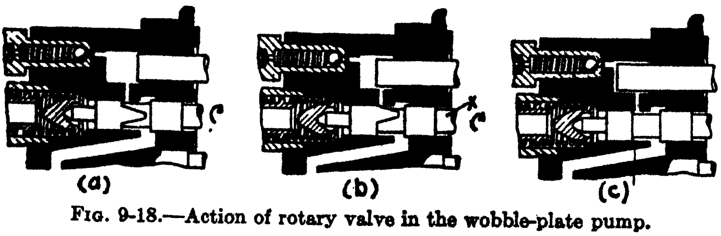


FIG. 9-17.—Multicylinder wobble-plate injection pump. (*Courtesy of Ex-Cell-O Company.*)



(a) (b) (c)

FIG. 9-18.—Action of rotary valve in the wobble-plate pump.

is reduced, and vice versa. Since one land *X* controls the injection duration of all pumps, the fuel distribution to each cylinder is the same. The outward movement of the other governor weight *K* pushes the sleeve *Q* to the left, and because of the spiral spline *N*, the timing of the land *X* is advanced, giving the same advantages as an automatic spark advance on gasoline engines. Because of the spline *V*, the axial movement of *Q* does not affect the axial position of *W* and *X*.

At the bottom of the unit is located a fuel-transfer pump. Plunger *T* is driven through a rocker arm *U* from the push rod of the bottom pump. Plunger *S* may be operated manually by the handle *h* for priming or filling the injection system. A small automatic pump contained in the plunger *T* picks up oil leakage from the pumps.

A De La Vergne cam-operated pump is shown in Fig. 9-19. A tappet *c* operates the lapped-in main plunger *p*. Near the end of the injection stroke, *c* begins to push upward the guided push rod *a* and, by means of levers *h*, *g*, and the small packingless plunger *d*, opens the pressure-relief valve *e*, thus terminating the fuel delivery. The governor changes the timing of the by-passing by raising or lowering the fulcrum point *h*.

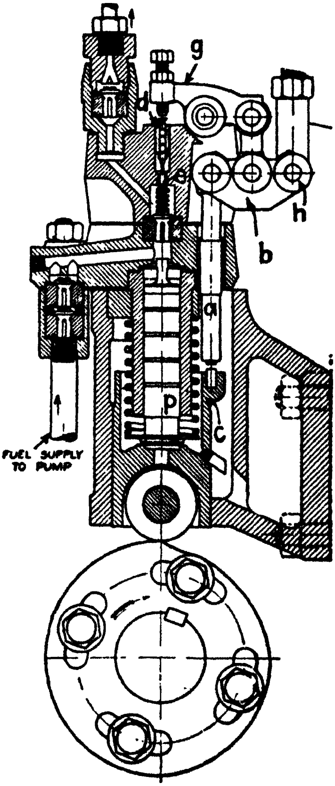


FIG. 9-19.—De La Vergne fuel-injection pump.

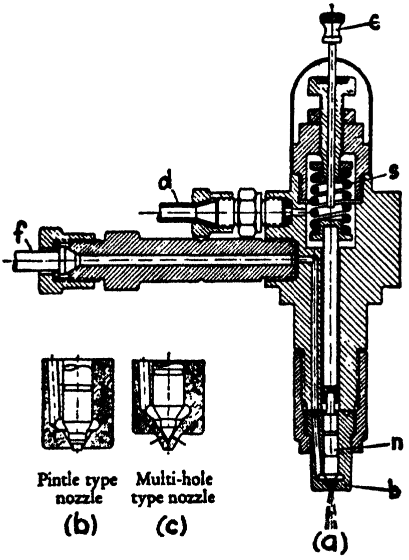
Injection Nozzles.—These nozzles can be divided also into two main types with certain subtypes:

1. Closed nozzles: (*a*) single-hole, (*b*) multihole, and (*c*) pintle type.
2. Open nozzles: (*a*) single-hole and (*b*) multihole type.

The single-hole nozzles have a good penetration but require a higher injection pressure in order to obtain a fine atomization.

The multihole nozzles give a good atomization but are more easily clogged up by foreign particles. They are made with from five to eight holes, from 0.006 to 0.020 in. in diameter. The pintle nozzle can be designed to produce various spray-cone angles.

At present most engines have individual pumps and hydraulically operated injection nozzles, which have a spring-loaded valve with a differential stem, Fig. 9-20: *f* is the fuel-oil connection and *d* the drain; the fuel valve *n* is opened by the oil pressure on the lower ring area of *n*, when the pump on its delivery stroke builds up a sufficient pressure; this pressure is regulated by adjusting the spring *s*. To decrease friction and obtain a snappier opening and closing of the valve *n*, the latter is made packingless and is lapped-in in the body *b*; both *n* and *b* are made of nitralloy steel and are heat-treated before lapping-in, which gives corrosion-resisting glass-hard surfaces, reducing the wear to a minimum if the oil is clean. Figure 9-20*a* shows a fuel-nozzle holder with a single-hole nozzle; Fig. 9-20*b* shows a pintle-type nozzle tip, and Fig. 9-20*c* shows a multihole nozzle tip.



Pintle type nozzle (b) Multi-hole type nozzle (c) (a)

FIG. 9-20.—Hydraulically operated fuel nozzles.

Example 9-6.—The diameters of the spring-loaded valve, Fig. 9-20*a*, are $d_1 = 0.200$ in., $d_2 = 0.110$ in.; the valve lift is 0.020 in. Other data are as given in Example 9-4. Determine the required spring force with the valve seated, the spring scale, and the oil pressure, when the valve closes; assume a gas pressure of 650 psi at that moment.

Diameter, in.	Area, sq in.	Pressure, psi	Force, lb
0.200	0.031416	1700	53.41
0.110	0.009503	1700	−16.16
0.019	0.000283	440	0.12
			Net force 37.37

While the valve stays open, the full pressure will act against its whole area; therefore the spring scale must be

$$F_0 = \frac{53.41 - 37.37}{0.020} = 809 \text{ lb/in.}$$

Just before closing, the condition of equilibrium will be

$$(0.031416 - 0.000283)p + 0.000283 \times 650 = 37.37 \text{ lb}$$

$$p = (37.37 - 0.18)/0.03112 = 1193 \text{ psi.}$$

Figure 9-21 shows an automatic single-hole fuel nozzle used in Benz's precombustion engines. The atomization is improved by the fuel streams impinging one upon another when they meet in the center. The lower part of the precombustion chamber has a number of holes *h*, which also help to mix the unburned fuel from the chamber with the air in the main combustion space.

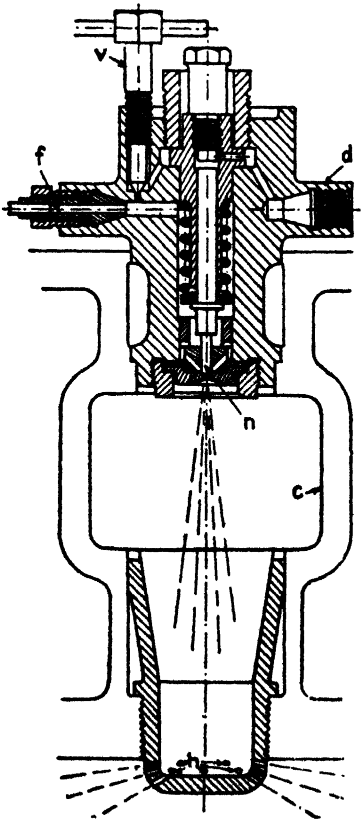


FIG. 9-21.—Benz precombustion-chamber and fuel nozzle.

Open-type nozzles have the fuel orifice, or orifices, and parts of the fuel passageway open to the cylinder pressures at all times. One or more check valves are used to stop the flow of cylinder gases into the injection system and to prevent the dribbling of fuel after the injection is com pleted. To improve atomization and distribution, the. fuel usually is given a whirling motion, before it passes through the orifices.

The injection of fuel from an open-type nozzle begins when the pump raises the pressure sufficiently to open the check valves and terminates when the fuel pressure drops below the combustion-chamber pressure. Thus, the pressure difference which causes the fuel flow is low at the beginning and end of the injection, and poor atomization occurs under these conditions.

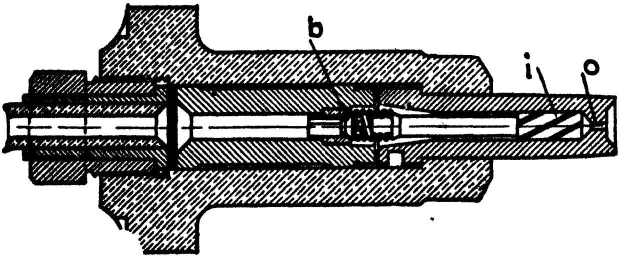
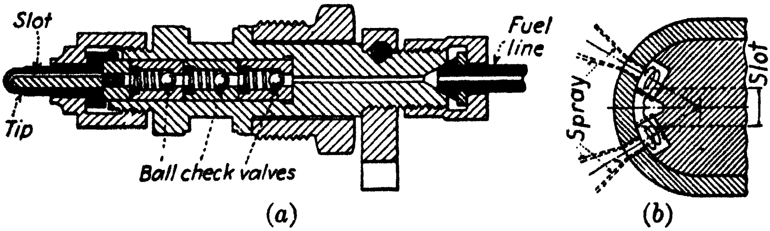


FIG. 9-22.—De La Vergne fuel nozzle.

Figure 9-22 shows a single-hole De la Vergne nozzle with a spring-loaded check valve *b*, an atomizer formed by the insert *i* with spiral grooves for better distribution of the fuel, and a long orifice *o* for good penetration.



(*a*) (*b*)

FIG. 9-23.—Open-type injection nozzle.

Figure 9-23*a* shows a Hesselman multihole nozzle; and Fig. 9-23*b* shows the nozzle tip, in which a longitudinal slot, connected tangentially with recesses behind the orifices, produces a whirling motion in the spray.

Constant-pressure Injection. *Pumps.*—The pump delivers a constant amount of fuel, slightly in excess of the maximum requirement: and the surplus is discharged back to the fuel tank through a spring-loaded relief valve, called *unloader* or *pressure regulator*. The pump motion has no relation to the piston position and may be operated from either the camshaft or crankshaft.

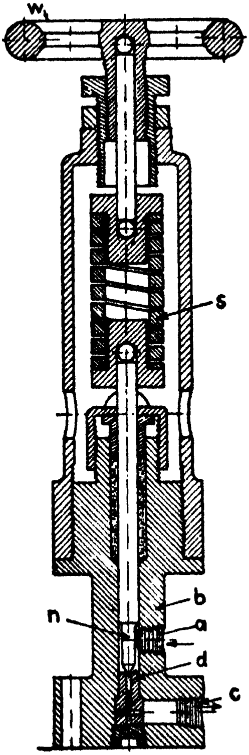


FIG. 9-24.—Unloader valve.

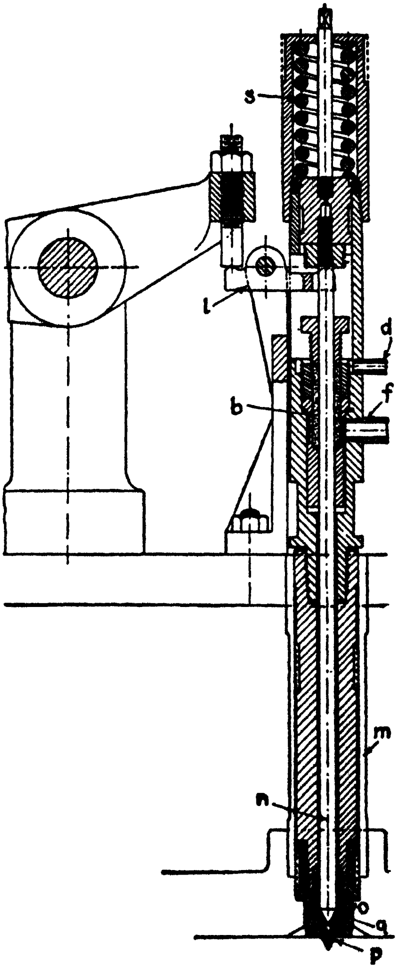


FIG. 9-25.—Western-Enterprise mechanical-injection fuel nozzle.

Smaller engines have a single-plunger pump, irrespective of the number of engine cylinders; larger engines have pumps with several plungers. The pressure maintained at normal load and speed is from 3500 to 4500 psi. The pumps are of the packingless type and are driven by cams or eccentrics.

Unloader.—The unloader body *b*, Fig. 9-24, is connected at *a* to the high-pressure fuel line and at *c* to a discharge pipe leading to the fuel tank; the small opening in a hardened-steel plug *d* is closed by the needle valve *n* loaded by a stiff spring *s*. When the pressure in the fuel line rises above the limit as set by the spring *s*, the needle *n* is lifted and the excess oil is discharged through *c*. The handwheel *w* serves to change the compression of the spring *s* according to the pressure which is desired.

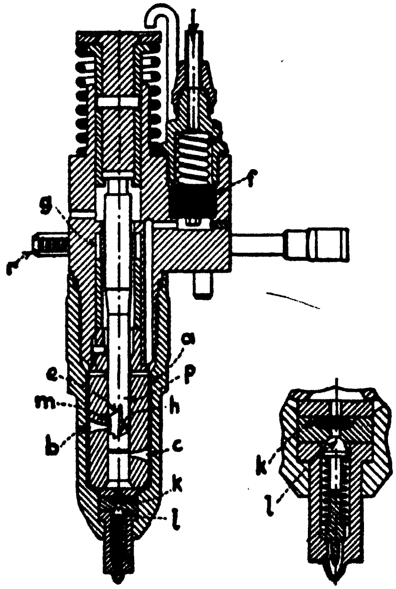


FIG. 9-26.—General Motors unit fuel injector.

Injection Nozzle.—Figure 9-25 shows a mechanically operated nozzle: the nozzle tip *p* has several small holes whose center lines form angles of about 60° with the axis of the needle valve *n*; *f* is the high-pressure fuel-oil connection and *d* the drain pipe for oil, which may leak through the packing box *b*. The intermediate lever *l* permits the use of the more convenient push rods, instead of pull rods, with a low-located camshaft.

Unit Injectors.—Unit injectors combine the pump and injection nozzle and eliminate the high-pressure pipe connection which, under certain conditions, creates pressure waves and thus disturbs the fuel discharge.

The plunger *p*, Fig. 9-26, has a center passage extending from the bottom of *p* to the metering recess *m*. The upward motion of the plunger permits fuel to flow from the fuel cavity *a* through the upper port *b* and plunger passage, as well as through the lower port *c* into the pump barrel below. During the first part of the downward stroke of *p*, the fuel returns to the cavity *a* until *p* covers the port *c*, and later the upper helix *e* covers port *b*. Turning of *p* by means of a rack *r* and gear *g* changes the position of the helix and thus varies the quantity of fuel under the plunger at the beginning of the effective stroke and the timing of the injection period as well.

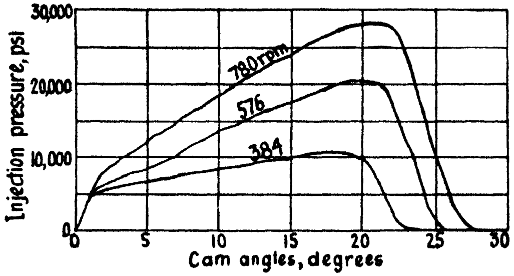


FIG. 9-27.—Pressures in a unit injector.

The plunger motion builds up a fuel pressure that opens the check valves *k* and *l* and forces the fuel through the small orifices in the open-type nozzle. The injection ends when the lower helix *h* uncovers the port *c* and thus relieves the fuel pressure. A filter unit *f* assures that only perfectly clean fuel will reach the small orifices.

With a unit injector the injection velocity through the orifices is proportional to the plunger speed, which itself is proportional to the engine speed. On the other hand, the injection velocity, according to equation (9-8), is proportional to $\sqrt{p_1 - p_2}$; thus with p_2 constant, the injection pressure p_1 increases as the square of engine speed. However, tests, Fig. 9-27, show that the pressure does not increase as fast, which is due to compressibility of the fuel, elasticity of metal parts, and leakage of the fuel.

Another unit injector is shown on Fig. 9-28: The fuel is delivered by a low-pressure pump past the ball check valve *v*; during the end of the suction and most of the compression stroke, the plunger *n* moves upward, creating a vacuum which prevents the fuel charge from oozing down into the combustion space. At the moment of injection, plunger *n* moves down rapidly and injects the fuel, which by that time is preheated and partially evaporated.

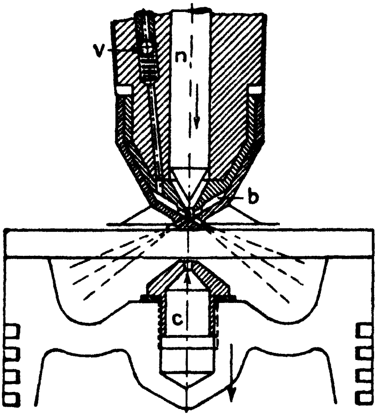


FIG. 9-28.—Cummins injection nozzle and turbulence chamber.

9-5. Problems.—**1.** Enumerate various methods of fuel injection, stating for each its advantages and disadvantages.

2. Determine the weight and volume of fuel per cylinder per cycle in a five-cylinder four-stroke air-injection engine developing 1000 hp at 200 rpm and having a fuel rate of 0.395 lb/hp-hr; the injection-air pressure is 1050 psig.

3. Determine the bore and stroke of the pumps for the engine of problem 2; the volume of the clearance space in the pump is 7.5 times larger than the effective piston displacement; the pipe connecting the pump to the fuel nozzle has a $\frac{5}{16}$-in. diameter and is 38 in. long.

4. Determine the velocity of the injection air entering the cylinder of a diesel engine if the compression pressure is 525 psig, and the air pressure in the injection nozzle is: (*a*) 900 psi, (*b*) 1000 psi, or (*c*) 1100 psi; the air is cooled to room temperature.

5. Determine the velocity of the injection air entering the cylinder of a two-stroke air-injection engine with a compression pressure of 510 psig if the air pressure in the injection nozzle is: (*a*) 850 psi, (*b*) 950 psi, or (*c*) 1050 psi.

6. Work problem 4, cases (*a*) and (*b*), assuming that the air temperature is 150 F.

7. Work problem 5, case (*b*), for air temperatures of 70, 140, and 210 F; present the results by a curve.

8. Determine the weight and volume of fuel per cycle, per fuel nozzle, in an eight-cylinder 7500-hp two-stroke-cycle double-acting marine engine operating at 100 rpm and having a fuel rate of 0.40 lb/hp-hr; 22° API oil is used; two fuel nozzles are used in each combustion space.

9. Determine the diameter of the single-hole fuel orifice for the engine in problem 8; the period of injection is 36°, the oil pressure is 2800 psig, the compression pressure is 400 psi, and the maximum pressure is 520 psi.

10. Determine the velocity of injection for data of problem 9.

11. Determine the weight and volume of fuel per cylinder per cycle in a six-cylinder 300-hp four-stroke engine operating at 1200 rpm and having a fuel rate of 0.41 lb/hp-hr; 35° API oil is used; the injection pressure is 1650 psi.

12. Determine the diameter of the fuel orifice for the engine of problem 11 if the injection duration is 34° of the crankshaft travel: (*a*) for a single-hole nozzle and (*b*) for a five-hole nozzle.

13. Determine the velocity of injection for data of problem 12.

14. Determine the bore and stroke of a pump for the engine discussed in problems 8 and 9; assume that the volume of the fuel inside the pump is nine times the effective piston displacement and that the pipe connecting each pump and its fuel nozzle is ¼ in. in diameter and 36 in. long.

15. Determine the mean oil velocity in the high-pressure fuel line of problem 14.

16. Determine: (*a*) the power required to drive each pump of problem 14, (*b*) the total power requirement of the whole injection system of the engine and (*c*) its percentage compared with the power developed by the engine; assume a leakage loss of oil of 2.5 per cent and a mechanical efficiency of the drive mechanism of 75 per cent.

17. Determine the bore and stroke of a pump for the engine discussed in problems 11 and 12; assume that the volume of the fuel inside the pump is eleven times the effective piston displacement and the pipe connecting the pump to the fuel nozzle is ¼ in. in diameter and 22 in. long.

18. Determine the mean oil velocity in the high-pressure fuel line of problem 17.

19. Determine: (*a*) the power required to drive the pump of problem 17; (*b*) the percentage of power required, referred to the rated engine power; assume a leakage loss of 3 per cent and a mechanical efficiency of 77 per cent.

20. The diameters of a spring-loaded valve, Fig. 9-20*a*, are $d_1 = 5$ mm; $d_2 = 3$ mm; $a = 0.45$ mm, the lift is 0.5 mm; the oil pressure to open the valve is 1,650 psig; the pressure in the cylinder at the beginning of the injection is 450 psi, at the end of injection is 600 psi. Determine the required spring forces with the valve seated and open; determine the oil pressure when the valve closes.

21. The diameters of a pintle-type valve, Fig. 9-20*b*, are $d_1 = 5$ mm, $d_2 = 3$ mm, the fuel orifice $d = 2.5$ mm, and the diameter of the pintle 1.5 mm; the valve lift is 0.25 mm; the injection pressure is 1750 psig; the pressure in the cylinder is 470 psi at the beginning of the injection and 750 psi at the end. Determine the required spring forces with the valve seated and open, the scale of the spring, and the oil pressure when the valve closes.

22. For the valve and its spring referred to in problem 21, find the spring forces and the closing oil pressure if the injection pressure is raised to 2000 psig, all other data unchanged.

CHAPTER 10

COMBUSTION IN COMPRESSION-IGNITION ENGINES

10-1. Combustion Phenomena. *Air-injection Engines.*—The phenomena of ignition and of combustion are basically the same in air- and airless-injection engines. However, the atomization and distribution of the fuel in air-injection engines is so efficient, and without any appreciable ignition lag, that no particular problems concerning combustion arose until airless injection became more widely accepted and gradually superseded air injection. The brief analysis presented in conjunction with Fig. 9-4 gives a sufficient account of air-injection combustion. The following discussion refers to airless-injection engines.

Ignition.—When injection begins, finely atomized, relatively cold fuel-oil particles come in contact with air preheated by the compression stroke. First, their temperature increases; then they begin to vaporize, and the temperature of the vapor particles increases; a low oxidation reaction starts, until the temperature reaches the ignition temperature, when a quick chemical reaction causing an increase of pressure and temperature starts and spreads to the rest of the fuel in the combustion space. Contrary to ignition by a spark plug, ignition by compression starts not at some definite point, but at a place determined by conditions of temperature and fuel distribution and may even start at several points, either simultaneously or consecutively.

Combustion.—While a certain minute amount of the fuel must be vaporized, in order to ignite the fuel charge, the subsequent burning of the rest of the charge, according to information gained with high-speed engines, takes place from the surface of the liquid droplets when the latter are brought in contact with oxygen. The necessary prerequisites are that the droplets be heated to the temperature of ignition at the existing pressure and that the products of combustion be scoured away as soon as they are formed in order to bring the fuel particles again in contact with molecules of oxygen. Thus an essential condition for efficient combustion, particularly in high-speed engines, is a sufficient relative motion between the fuel droplets and the air. This condition can be met by rapid movement either of the fuel or of the air, or of both, provided that their direction is different.

On the basis of research data, several important general facts about combustion in airless-injection compression-ignition engines have been established. First, the combustion process, when it is started by compression ignition, is considerably more rapid than by spark ignition. This is the result of the higher temperature and density of the charge when combustion starts. Second, the rate of burning does not depend upon the fuel used. Third, the ignition quality of a fuel, its cetane number, affects the smoothness of running of an engine but does not appreciably affect either the power output or the efficiency of an engine. Thus, the problem of obtaining a satisfactory engine performance consists chiefly in adjusting conditions so as to keep down to certain limits the final pressures, the rate of rise of the combustion pressure, and the noise, to obtain a steady combustion process at a constant load, to maintain this steadiness at varying loads and speeds, and, finally, to assure uniform combustion in each cylinder of a multicylinder engine.

A solution of these problems becomes possible only with a clear understanding of the details of combustion and of all factors which influence it.

10-2. Combustion Stages.—Modifying Ricardo's original analysis, the combustion process in compression-ignition engines, particularly of the high-speed type, is at present believed to take place in *four* separate stages. During the first period there is no visible pressure rise; hence it is called the *delay period.* Fuel is constantly coming in through the nozzle, and when ignition finally occurs, there is already in the combustion chamber an appreciable amount of fuel, finely divided and evaporated, ready for combustion. This fuel begins to burn very rapidly because of the rise of temperature in the combustion chamber and possibly because of the existence of several points of ignition. This period of *rapid combustion* is accompanied by a rapid pressure rise and forms the second stage of the process. After the period of rapid combustion, the fuel which has not yet burned and the fuel subsequently injected burn at a rate that depends upon the amount and distribution of oxygen left in the air charge. This period is the third stage or *gradual*, or *controlled, combustion*. The fourth stage, or *afterburning*, is burning of the fuel after injection terminates. This stage cannot be controlled and is very undesirable as its efficiency is comparatively low.

Figure 10-1 illustrates the process by a pressure-time diagram: the fuel is injected at point 1, but ignition does not start until point 2. Angle *a* represents the delay period also called *ignition lag*. For a certain engine the delay period depends upon many factors, as will be shown later. From point 2 to point 3, corresponding to a crank angle *b*, the flame spreads from the initial nucleus to the main body of the fuel charge. Similar to the conditions of spark-ignition engines, the flame velocity and the pressure rise depend upon turbulence; this second phase of the process is very important for a smooth operation of an engine. During the third stage, crank angle *c*, from point 3 to 4, the fuel burns as it leaves the fuel nozzle and the pressure may either increase, remain constant, or decrease, depending upon the rate at which it is delivered to the combustion chamber. The angle *c* is a function of the load which the engine is carrying. The fourth stage, afterburning, is not apparent on the indicator diagram. On a diagram taken from an engine, the dividing points 2, 3, and 4 are not pronounced, and the four stages merge one into another gradually.

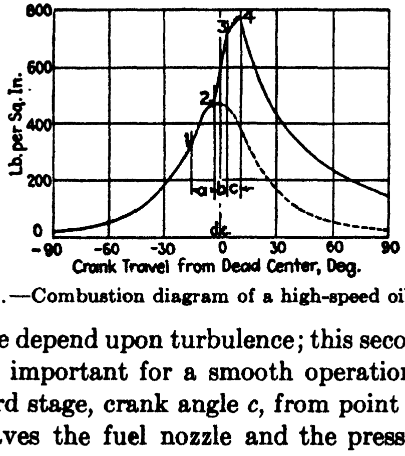


FIG. 10-1.—Combustion diagram of a high-speed oil engine.

10-3. Delay Period.—This period itself is made up of two parts: a heating period when the cold fuel droplets are heated, vaporized, and brought up to their ignition temperature and a period of true ignition delay that ends when the first particles actually ignite. However, in engines it is difficult to distinguish between these two periods, so the delay period is measured from the beginning of the injection to the moment of ignition

The delay period presents a great interest, as it materially influences the operation of an engine: A shorter delay period gives a smoother operation; a longer delay period results in a rougher and noisier running engine. The factors that influence the length of the delay period are:

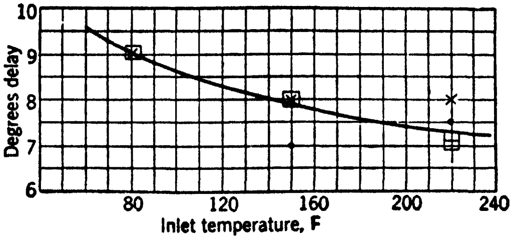


FIG. 10-2.—Effect of inlet temperature on delay angle.

(*a*) temperature of the air charge,
(*b*) pressure of the charge,
(*c*) atomization of the fuel,
(*d*) timing of injection,
(*e*) engine speed, and finally,
(*f*) ignition quality of the fuel, its cetane number.

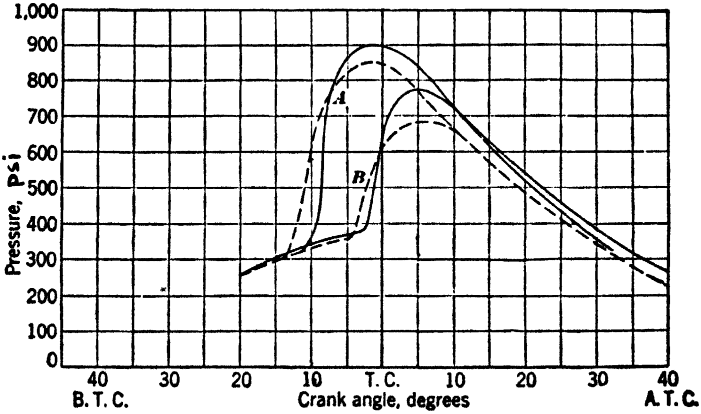


FIG. 10-3.—Effect of injection timing and jacket temperature on pressure curves. Curves *A* and *B*—injection starts 20 and 10° B.T.C., respectively; solid and broken-line curves—jacket temperature 150 and 300F, respectively.

Temperature.—Figure 10-2 shows the typical effect of inlet temperature on the delay angle of an experimental engine.[1] The same condition is confirmed by another engine, Fig. 10-7.

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[1] ROTHROCK, A. M., *SAE Journal*, vol. 34, June, 1934.

Figure 10-3 shows the effect of the jacket temperature on the delay angle.[1] In both cases the increase of the respective temperature gives an increase of the temperature of the charge at the moment of injection, and the result is a decrease of the delay angle with an increase of the charge temperature.

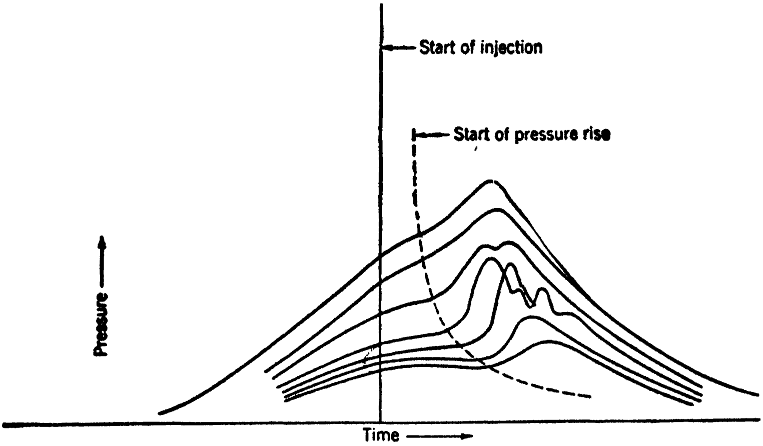

FIG. 10-4.—Effect of pressure on ignition lag, inlet throttled.

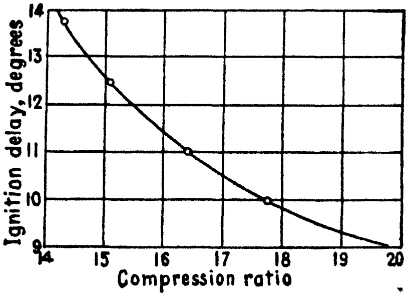

FIG. 10-5.—Effect of compression ratio on ignition delay.

Pressure.—The influence of the pressure of the charge due to an increase of the initial pressure is shown in Fig. 10-4.[2] An increase of the compression ratio has the same effect, Fig. 10-5.[3] However, in this case the delay angle is influenced also by the temperature of the charge, which goes up with an increase of the compression ratio.

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[1] ROTHROCK, A. M., and WALDRON, C. D., *NACA Rept.* 525, 1935, p. 4.

[2] BOERLAGE, G. D., and BROEZE, J. V., *SAE Journal*, vol. 31, July, 1932.

[3] POPE. A. W., JR., and MURDOCK, J. A., *SAE Journal*, vol. 30, March, 1932.

Tests indicate that the influence of the temperature T, °R, and pressure p, psia, of the air in the combustion chamber at the time of fuel injection are more pronounced than all other factors and, at the same time, are so definite that the delay period z in seconds can be presented by the equation[1]

$$z = 0.0104e^{8200/T}/p^{1.19} \tag{10-1}$$

This expression is reasonably accurate for fuels with a cetane number of 40 and higher.

Atomization evidently has a great influence upon the delay period—finer atomization, smaller droplets, decreases the delay period. Therefore as fine an atomization as possible, consistent with other requirements, penetration and distribution, is the aim of every fuel-nozzle design. However, it is hardly possible to give any definite numerical or graphical relation between the degree of atomization and the delay period, except that with a very fine atomization the delay period may be cut down to about 0.0008 sec, and with a coarse atomization it may be 0.005 sec.

Timing.—The influence of the timing of the injection can be seen from Fig. 10-3: The minimum delay angle is obtained with a certain timing of the injection, in this case about 11.5° before dead center. Both advancing and retarding of the injection increase the delay angle.

Engine-speed influence for an experimental engine using 40 cetane fuel is shown in Fig. 10-6 curve 1;[2] curves 2 and 3, for other engines and different fuels, show the same trend.[3] With an increase of the engine speed, the delay angle increases, although very little; however, the delay period, seconds, decreases rather rapidly. In other engines and with other fuels the delay angle may increase faster with an increase of the engine speed and in some cases, on the contrary, may even decrease. However, the delay period expressed in seconds, with any fuel, always decreases with the engine speed. This is caused by an increase of the air turbulence, which in turn assists the heating of the fuel particles. Chiefly because of this condition, compression-ignition engines are able to run at those high speeds at which some of them operate at present.

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[1] *Sulzer Tech. Rev.* No. 3, 1939, p. 8.

[2] RICARDO, H R., and GLYDE, H. S., "The High-speed Internal-combustion Engine," p. 390, Interscience Publishers, Inc., New York, 1941.

[3] ZINNER, KARL, Neuere Anschauungen über den Zündvorgang im Dieselmotor, *Z. Ver. deut. Ing.*, vol. 83, No. 39, p. 1077, Sept. 30, 1939.

Ignition Quality of Fuel.—The influence of the ignition quality of the fuel is very pronounced, as illustrated by the typical

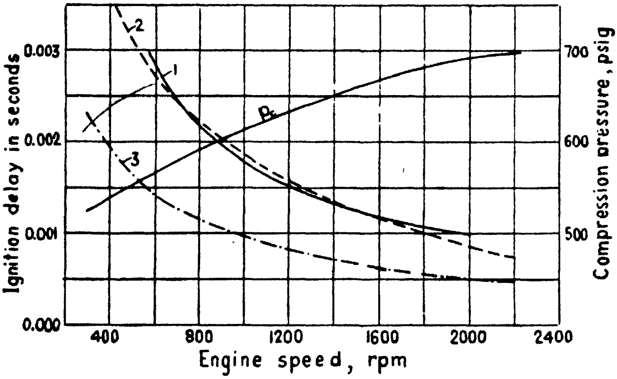

Fig. 10-6.—Effect of engine speed on compression pressure and ignition delay.

Fig. 10-7.[1] By using a fuel with a higher cetane number, all other conditions being the same, the delay angle could be reduced from 18 to 8°, and the time element of delay from 0.0020 to 0.00089 sec.

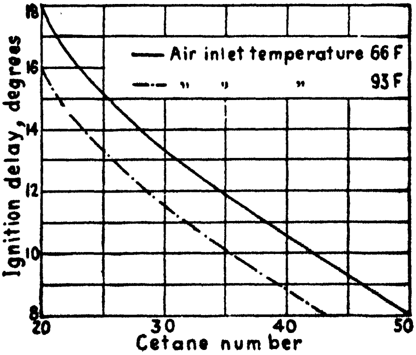

Fig. 10-7.—Effect of cetane number on ignition delay.

Unfortunately, unlike spark-ignition engines, in which a fuel with a higher octane number improves the performance of an engine in every respect, a fuel with a high cetane number improves the performance of the engine only in respect to fuel knock. A high cetane fuel does not give any higher power output. On the contrary, by accelerating combustion, it prevents good fuel distribution and requires a higher air excess.[1,2]

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[1] Ricardo and Glyde, *op. cit.*, p. 391.

10-4. Second Stage.—For a smooth, noiseless operation of an engine the second stage of combustion is very important, because the rate of pressure rise, per degree of crank travel, must not exceed a certain limit. The actual limit depends upon certain construction features of the engine, but in most engines should not exceed 30 to 35 psi/degree.

A direct control of the burning during this stage is almost impossible; the indirect control consists in reducing the amount of fuel awaiting combustion, and this is obtained by reducing the delay period. However, this second phase is important, as it is responsible for the so-called *fuel knock.* It may present a problem in engines with air stagnation in the combustion chamber, but in the presence of a sufficient air turbulence and a small delay period, it usually takes care of itself.

Detonation.—Strictly speaking the term *detonation*, in reference to compression-ignition engines, is not correct. The phenomenon, which resembles the detonation that occurs in spark-ignition engines, is an excessively high rate of pressure rise during the second phase of combustion, accompanied by a more or less pronounced knocking sound; a more appropriate name for it is *fuel knock*. As already explained, the cause of this knocking is the presence of an excessively large portion of the fuel in the combustion chamber when ignition starts. This may be caused either by an excessively large delay period or by a too fast rate of fuel injection. The methods to eliminate fuel knock are those which were discussed in conjunction with the decrease of the delay period. Often the lowering of the injection rate is not feasible without affecting adversely the main combustion.

10-5. Third Stage.—This stage begins after the pressure and temperature in the combustion chamber have been raised by the second stage to the height that ensures the burning of the fuel as it is delivered from the fuel injector. This phase should form, both qualitatively and quantitatively, the main part of the combustion process, the first and second phases being only an

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[1] ROTHROCK, A. M., *Diesel Power*, vol. 14, p. 31, January, 1936.

[2] BOERLAGE, G. D., Detonation and Autoignition, *NACA Tech. Mem.* 843, 1937.

introduction to it. Qualitatively the third phase is very important, because during it there exists a direct control over the rate of burning and hence over the combustion pressures. The rate of burning, in turn, depends upon the rate of injection, fineness of atomization, and uniformity of distribution of the fuel in the combustion chamber. The distribution depends upon the number and direction of the nozzle orifices and is materially assisted by a directed turbulence in the air charge.

The present trend in engine design is to obtain not only the highest possible efficiency but also the highest power output from a given cylinder size. As stated before, the power output increases with a decrease of the excess air required for complete combustion; and, on the other hand, an efficient combustion with a small excess air is possible only if a high relative motion is created between the fuel globules and the air. The problem is clearly very important, but the solution is not simple. This explains the variety of methods used in different engine designs.

These methods can be classified into the following groups:

1. Use of high fuel-injection pressures to impart to the fuel particles a high velocity.
2. Use of a unidirectional high-velocity flow of air and injection of the fuel across the air stream.
3. Use of a so-called *precombustion chamber*, in which the pressure rise of the second phase of combustion is utilized to impart a high velocity to the remainder of the fuel charge and to create strong turbulences in the air charge.
4. Creating an air flow into the combustion chamber from a so-called *air cell*.
5. Creating an additional turbulence by means of a so-called *energy cell*.

High Injection Pressures.—This method is handicapped by the fact that the velocity which a liquid jet can gain due to a pressure difference when leaving an orifice is limited; if the pressure difference exceeds a certain value, a further increase will not increase the velocity but will result in a finer atomization. Furthermore, the velocity gained by a high pressure difference is soon dissipated in a high-density air charge. Finally, very high pressures, due to the compressibility of the fuel oil and the elasticity of the pipe lines, add considerably to the problem of metering and timing.

The other four methods will be discussed in conjunction with the design of combustion chambers.

10-6. Fourth Stage.—This last and undesirable stage of combustion after injection is completed is influenced by two factors, excess air and turbulence. Since a greater excess air means lower power output, the better method of shortening afterburning is by increasing the turbulence in the combustion chamber, toward the end of the combustion process. This may be accomplished to a certain extent by the use of the above-mentioned energy cell. However, the best procedure is to eliminate this stage as much as possible by effectively controlled burning in the third stage.

10-7. Problems.—**1.** Describe briefly the phenomenon of ignition of the fuel charge in a compression-ignition engine.

2. Describe briefly the phenomenon of combustion in a compression-ignition engine.

3. Explain what is meant by the four stages of combustion.

4. Which of the two stages, the first or second, is more important for a satisfactory operation of a compression-ignition engine? Explain why.

5. Enumerate the factors which affect the length of the delay period and state which ones affect it the most.

6. Compute the expected delay period and the delay angle for an engine with a compression ratio $r = 13$, running at 900 rpm, initial compression temperature 130 F; exponent of the compression line $n = 1.39$.

7. Compute the expected delay period and the delay angle for an engine with a compression ratio $r = 15$, operating at 1200 rpm; assume the exponent of the compression line $n = 1.39$; compare two conditions, when the final suction temperature is (*a*) 125 F and (*b*) 135 F.

8. Explain the influence of engine speed upon the ignition delay.

9. Compute the delay angles for the three curves given in Fig. 10-6, and replot these curves, using delay angles for ordinates.

10. Explain the cause of fuel knock in compression-ignition engines.

11. Enumerate the methods which are used to obtain efficient combustion with a relatively small excess air.

12. Explain the qualitative and quantitative importance of the third stage of combustion in compression-ignition engines.

13. Enumerate and explain the handicaps for the use of high injection pressures, with the object of assisting the third stage of combustion in compression-ignition engines.

CHAPTER 11

COMPRESSION-IGNITION COMBUSTION CHAMBERS

11-1. General Remarks.—The combustion chamber considered as a definite closed space is formed by three separate engine parts: the cylinder head, extension of the cylinder, or cylinder liner, and top of the piston.

In air-injection engines the atomization of the fuel and its mixing with the air charge are so efficient that no problem as to the best shape of the combustion chamber exists. The same is also true, to a certain extent, of the slow-speed medium-size airless-injection engines. Only with the advent of high-speed engines, with their smaller piston displacements and requirement of higher mean effective pressures, with better utilization of the air or smaller excess air, did the question of the best design of a combustion chamber arise and become of great importance. With this in mind, it may be stated that the two principal functions of the combustion chamber are, first, to assist and supplement the injection system in the preparation of the cylinder charge and, second, to obtain an efficient combustion of this charge. The combustion chamber and the injection system are so interdependent that a certain type of injection requires a certain combustion chamber, and vice versa. A combustion chamber and an injection system cannot be designed or considered separately.

11-2. Air Movements.—The importance of air movement, usually termed *turbulence*, has been discussed previously. Air movements help: (*a*) to distribute the fuel, (*b*) to mix it with air, (*c*) to assist combustion, and, finally, (*d*) to reduce afterburning.

Several types of air movements are distinguishable:

1. A *swirl*, or circular motion, imparted during the suction stroke or the scavenge period.
2. An *air turbulence* created by the piston during the compression stroke when a divided combustion chamber is used.
3. A *squish*, or flow of air, from the periphery to the center of the cylinder, created toward the end of the compression stroke.

4. *Turbulences* created during combustion by (*a*) precombustion chambers, (*b*) air cells, or (*c*) energy cells.

Suction swirl is induced when the air charge is admitted in a tangential direction and, being deflected by the cylinder wall, assumes a rotary motion about the axis of the cylinder. The tangential direction may be produced in one of several ways: (*a*) by masking one side of the inlet valve so that air is admitted only around a part of the periphery of the valve and in the desired direction, Fig. 11-1*a*; (*b*) by sloping the sides of the inlet part in the desired direction, Fig. 11-1*b*; or (*c*) by casting a lip *l*, Fig. 11-1*c*, over one side of the inlet-valve seat.

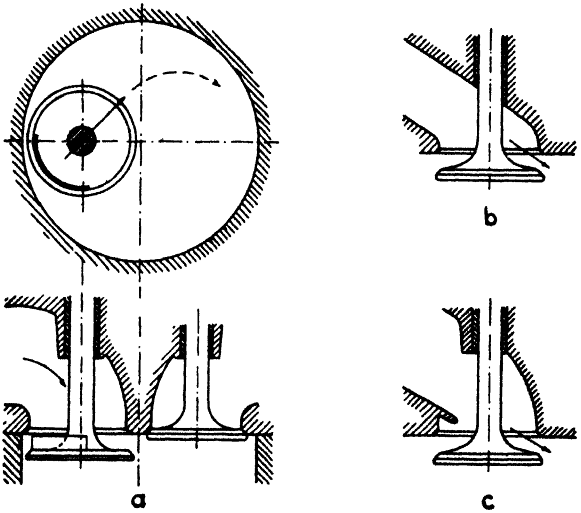

FIG. 11-1.—Methods of producing a suction swirl.

The masking of the inlet valve is the more common method, as it permits the finding of the best tangential direction by turning the valve about its center line. Its disadvantages are a certain restriction of the free passage area and the necessity of providing an arrangement to prevent rotation of the valve. However, experience shows that the effect of the area restriction is not very great.

The arrangements in Fig. 11-1*b* and *c* produce approximately the same restriction; and while their construction features are simpler, they do not have the flexibility of changing the direction of the air flow.

While the automobile spark-ignition sleeve-valve engine has practically disappeared now, sleeve-valve engines seem to present interesting possibilities as compression-ignition engines. In a double-sleeve engine with a reciprocating axial motion of the sleeves, an efficient swirl can be obtained by using tangential admission ports, Fig. 11-2*a*. In a single-sleeve construction the sleeve has both an axial and a rotary travel, and a swirling air motion is obtained even with radial ports, because admission starts at one corner of the port, Fig. 11-2*b*.

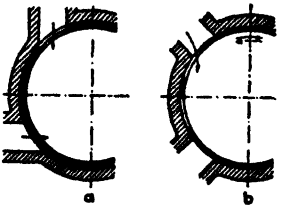

FIG. 11-2.—Air admission in sleeve-valve engines.

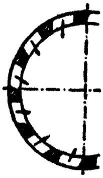

FIG. 11-3.—Scavenge-port arrangement for air swirl.

With two-stroke engines a tangential arrangement of the scavenge ports creates a strong air swirl, Fig. 11-3.

In conclusion, it may be stated that tests have shown that a swirl induced in the air charge during the suction or scavenging period is not reduced materially during the compression stroke and is still in effect during combustion.

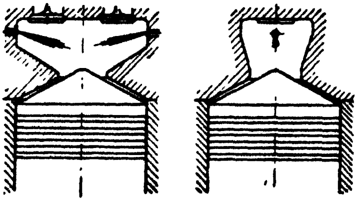

FIG. 11-4.—Price combustion chamber.

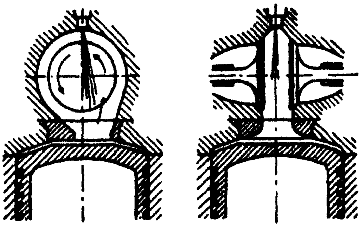

FIG. 11-5.—Vortex combustion chamber.

Compression turbulences are created if the combustion chamber proper is separated from the piston-swept cylinder volume by a restriction or throat, Fig. 11-4. Air is forced through it by the piston, particularly toward the end of the compression stroke. Again, as the piston starts the working stroke, there is an expansion back into the cylinder space past the corners of the throat, and the burning gases are subjected to further turbulence and better mixing.

By making the combustion chamber unsymmetrical, Figs. 11-5 and 30-10, a definite rotary motion, or vortex, can be created which is more efficient than a general, undirected turbulence. A still stronger turbulence, timed with the beginning of the fuel injection, can be obtained if the piston toward the end of the stroke gradually reduces the area of the throat. Figure 11-6 shows diagrammatically such a spherical combustion chamber and below, the forced-air-flow velocity through the throat into the chamber.

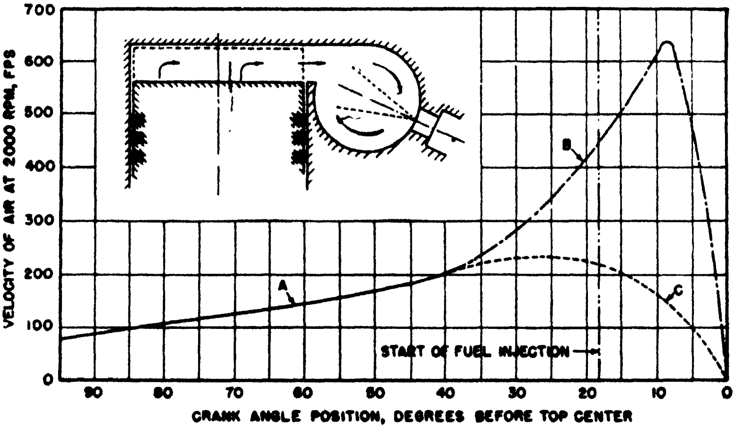

Fig. 11-6.—Hercules combustion chamber and air-flow velocity through the throat.

The absolute velocity naturally depends upon relative sizes of the piston area and the area of the throat and upon the piston speed, *i.e.*, engine speed. Divided combustion chambers require carefully proportioned throats: decreasing the throat gives a higher turbulence but causes difficult starting and considerable loss of heat to the walls; on the other hand, an excessively large throat is not effective in assisting combustion and gives poor low-speed performance.

Squish.—In engines with an open-type combustion chamber the squish is formed by a recess, located either in the piston, Fig. 11-7*a*, or, in some two-stroke engines, in the cylinder head, Fig. 11-7*b*. In order to force the air charge into this recess, the piston, at the dead-center position, is brought to within a very small distance of the cylinder head. This minimum distance or clearance varies with the engine size and design, but values as small as $\frac{1}{16}$ and $\frac{3}{64}$ in. are encountered. Even smaller clearances have been used, but these require special precautions in machining and operation.

Such a small clearance produces a radial flow of the air from the periphery to the recess when the piston approaches the dead-center position. This movement, termed *squish* by Ricardo, is added to any other turbulence that may already exist and is very helpful, since it is most vigorous at the dead center when it is most needed.

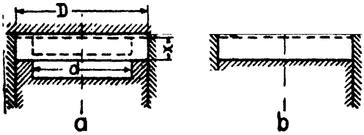

FIG. 11-7.—Combustion chambers with air squish.

The velocity of the air squish can be computed if the charge is considered as being divided into two volumes, the inner cylinder, whose volume is, using the designations of Fig. 11-7*a*,

$$V = 0.7854d^2(x + c) + V_1 \qquad (11\text{-}1)$$

where V_1 is the volume of the combustion recess in the piston or cylinder head, and the remainder in the outside ring-shaped volume.

The piston distance x from the dead center at a crank angle α is computed by the formula

$$x = r[1 - \cos\alpha + (r/2l)\sin^2\alpha] \qquad (11\text{-}2)$$

For certain cranks and piston positions the weights of the charge contained in the center part are found, and the differences will indicate how much of the charge has moved from the periphery to the center part. Computing the mean area A, through which this mass has flown, from the relation

$$A = \pi d[0.5(x_i + x_{i+1}) + c] \qquad (11\text{-}3)$$

and having found its mean specific weight permits finding of the corresponding mean squish velocity.

The following example will illustrate the procedure and show the magnitude of this squish velocity.

Example **11-1.—Find the squish velocity for a $4\frac{9}{16}$-in. × $5\frac{3}{4}$-in. engine which runs at 1600 rpm; the compression ratio is 16.0:1; the mechanical clearance between the piston and cylinder head $c = \frac{3}{64}$ in., Fig. 11-7*a*; the diameter of the recess $d = 2\frac{1}{2}$ in.; the temperature of the air charge at the**

end of the suction stroke is 140 F and the suction vacuum is 1.5 psi; connecting-rod length to crank-radius ratio is 4:1.

The piston displacement

$$V_d = 0.7854 \times 4.5625^2 \times 5.75 = 94.0 \text{ cu in.}$$

the total compression space

$$V_c = 94/(16 - 1) = 6.27 \text{ cu in.}$$

the clearance space on top

$$V_c = 0.7854 \times 4.5625^2 \times 0.0469 = 0.77 \text{ cu in.}$$

which leaves for the volume in the recess

$$V_r = 6.27 - 0.767 = 5.50 \text{ cu in.}$$

The investigation is made for every 5° of crank travel for the last 45°, column (1), Table 11-1.

The piston distances x from the dead center, column (2), were determined by expression (11-2). The height of the compression space corresponds to

$$5.75/(r - 1) = 5.75/(16 - 1) = 0.383 \text{ in.}$$

Therefore the compression ratios r, column (3), were computed from the expression

$$r = (5.75 + 0.383)/(x + 0.383)$$

The specific weights w, column (4), were determined by multiplying the initial specific weight 0.0594 lb/cu ft by the corresponding compression ratio.

The volumes of the charge in the center, column (5), were found by expression (11-1). Figures in column (6) were obtained by dividing those in column (5) by 1728 and multiplying by data from column (4).

Figures in column (7) are differences for data in column (6).

TABLE 11-1.—DETERMINATION OF SQUISH VELOCITIES

(1) Angle α deg to d. c.	(2) Piston distance x, in.	(3) Compression ratio r at x	(4) Sp. wt. w, $\frac{\text{lb}}{\text{cu ft}}$	(5) Volume in center V_2, cu in.	(6) Wt. of air in center, lb	(7) Wt. of air moved, lb	(8) Mean passage area, sq in.	(9) Squish veloc. v, fps
45	1.024	4.37	0.260	10.75	0.001617	0.000091	7.62	11.75
40	0.822	5.09	0.302	9.76	0.001708	0.000122	6.70	16.77
35	0.638	6.02	0.357	8.86	0.001830	0.000152	4.74	22.68
30	0.475	7.15	0.425	8.06	0.001982	0.000184	3.55	30.75
25	0.334	8.56	0.508	7.37	0.002166	0.000226	2.53	44.25
20	0.216	10.25	0.609	6.79	0.002392	0.000252	1.696	61.7
15	0.122	12.15	0.722	6.33	0.002644	0.000246	1.064	82.4
10	0.055	14.0	0.832	6.00	0.002890	0.000192	0.640	94.9
5	0.014	15.45	0.918	5.80	0.003082	0.000068	0.424	47.5
0	0.000	16.0	0.950	5.73	0.003150			

The data in column (8) were figured by expression (11-3) with values of x from column (2). Finally the mean squish velocity was found from figures of column (7), divided by the mean specific weight for the corresponding period, column (4), divided by the time element of one period,

$$t = \frac{60 \times 5}{1600 \times 360} = \frac{1}{1920} \text{ sec}$$

and divided by the mean area, column (8), itself divided by 144, to change it to square feet.

This example shows that the squish velocity attains a fairly high value, about 100 fps; and in spite of the decreasing piston speed, the maximum is very near the dead center, where it is most helpful.

In the above computations it was tacitly assumed that the density of the charge is the same throughout the combustion chamber. Actually the pressure and density will be higher at the periphery.

Pressure Head.—The pressure head, or pressure difference, which gives the squish velocity v can be computed from the equation

$$h = v^2/2g \tag{11-4}$$

This head is expressed, in feet, of the gas flowing and can be changed into pressure units, psi, by multiplying it by the mean specific weight of the gas w and by dividing it by 144,

$$\Delta p = (v^2/2g)w/144 = wv^2/9270 \tag{11-5}$$

Example 11-2.—Find the pressure differences for the engine in Example 11-1. Applying expression (11-5), it will be found that Δp is very small; it becomes measurable, $\Delta p = 0.12$ psi, first for the period from 25 to 20°; for the next period it becomes 0.27, then 0.57, and 0.85 psi, and for the last 5° period drops to 0.23 psi. The pressure head is very small, which is desirable.

The effectiveness of squish increases with a decrease of the clearance c and also of the ratio d/D. On the other hand, rounding off the edge of the recess or slightly increasing the mechanical clearance between the piston and cylinder head reduces the squish velocity and may ruin the performance of an engine.

Combustion Turbulences.—*Precombustion-chamber turbulence* is created when fuel, delivered to the precombustion chamber and assisted by the hot chamber walls, evaporates and partially burns in an insufficient amount of air. The ensuing sudden temperature and pressure rise causes the burning mixture to flow with a high velocity into the main, open part of the combustion space. The high velocity breaks up the coarser particles of the unburned fuel and creates a turbulence helpful for complete combustion.

Air-cell Turbulence.—The air is forced into an air cell during the compression stroke; when the piston moves away during the working stroke, it is discharged back into the main combustion space, creating a mild turbulence in the latter. Figures 11-12*a* and *b* show two air-cell types.

Energy-cell turbulence is produced not by air but by a stream of burned gases, as in a precombustion chamber. In a combustion chamber with an energy cell the fuel is injected into the main, open chamber in the direction of the cell. Part of the fuel, which is more thoroughly atomized, is dispersed in the open chamber; the more compact core of the fuel jet enters into the energy cell and begins to burn there; the resulting pressure rise blows the burning mixture back into the main chamber, creating a strong turbulence, which by the shape of the walls is transformed into a rotary swirl and helps to obtain complete combustion with a relatively small amount of excess air. The main disadvantage of this type of turbulence is that its intensity is affected by the relation between the volumes of the main chamber and the cell, by the size of the throat connecting them, by the shape of the fuel spray, and by the rate of fuel injection.

11-3. Combustion-chamber Types.—In the chronological order of their development, the various combustion chambers may be listed as follows:

(*a*) precombustion chambers,
(*b*) open, or quiescent, chambers,
(*c*) divided, or turbulent, combustion chambers,
(*d*) chambers with an air cell,
(*e*) chambers with an energy cell.

Precombustion chambers are built in various shapes and relative sizes; Fig. 9-21 shows one of the first successful designs, and Fig. 11-8 some latest American designs. Essentially a precombustion chamber is a quiescent chamber. The sudden rise of the pressure in the precombustion chamber during the first part of combustion throws the products of combustion and the unburned fuel into the main chamber and mixes it thoroughly with the remainder of the air either by dividing the flow into many streams, Fig. 9-21, or by creating a single stream of high velocity properly directed, Fig. 11-8. The piston top is made either flat or with a depression, Fig. 11-8*d*.

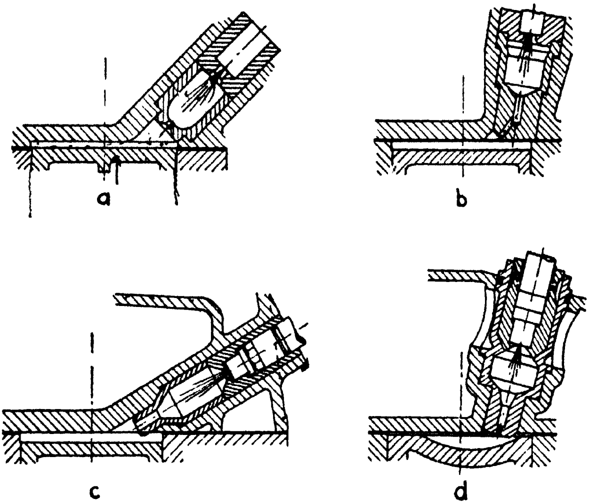

Fig. 11-8.—Precombustion chambers of American engines.

Open Chamber.—This term is applied to a combustion chamber in which all the air is confined in one undivided space. In four-stroke engines the chamber is basically a quiescent one, Fig. 11-9.

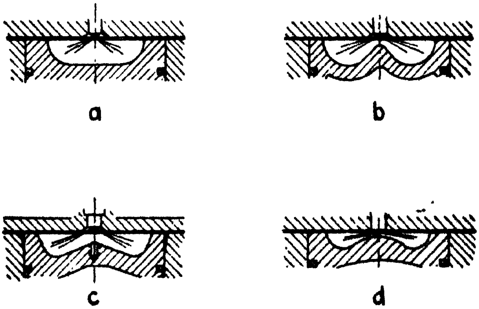

Fig. 11-9.—Open-type combustion chambers with air squish.

The distribution and mixing of the fuel with the air is effected chiefly by the fuel injector. However, these chambers can be used also with an induced suction swirl or with a compression air squish.

The good results obtained with open chambers of two-stroke engines, Fig. 11-10*a* and *b*, the latter showing the combustion chamber of a small opposed-piston engine, are due chiefly to the vigorous turbulence created by the swirling action of the tangential scavenging ports.

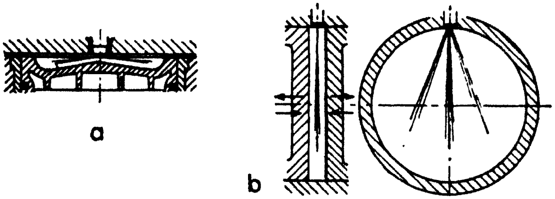

FIG. 11-10.—Open combustion chambers of two-stroke engines.

Divided chambers, of which two were shown in Fig. 11-4 and 11-5, are further illustrated in Fig. 11-11. They are used for engines from small to medium size and are equally effective with low and high rotative speeds.

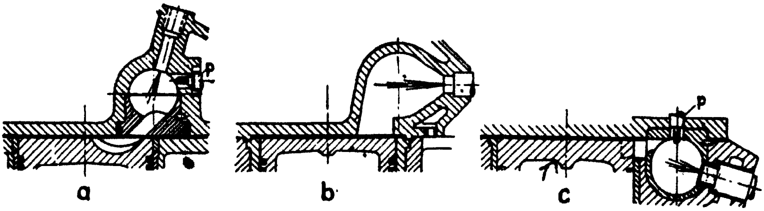

FIG. 11-11.—Turbulence combustion chambers.

Air-cell Chamber.—Two types are shown in Fig. 11-12. In some engines the air cell is located in the cylinder head. In either case the combustion turbulence is not quite strong enough and the maximum mep developed is not very high.

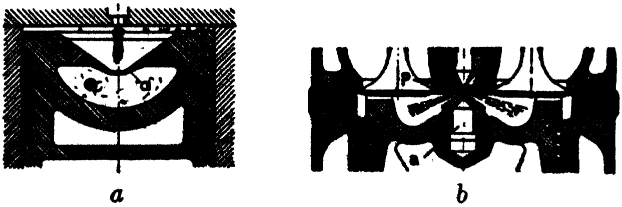

FIG. 11-12.—Combustion chambers with air cells.

Energy-cell Chambers.—**Figure 11-13*a* shows a so-called *double-lobe Lanova* combustion chamber used for small and medium-size**

cylinders; Figure 11-13*b* shows the *single-lobe* type adapted for engines with a very small engine bore. Figure 11-14 shows a combustion chamber with a Ramsey energy cell, which can be cut out by turning the valve *e* in order to increase the compression ratio for easier engine starting.

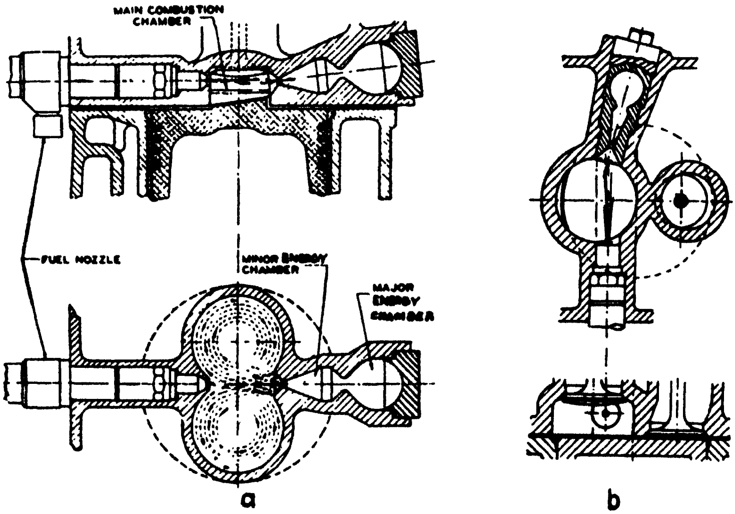

Fig. 11-13.—Combustion chambers with Lanova energy cell.

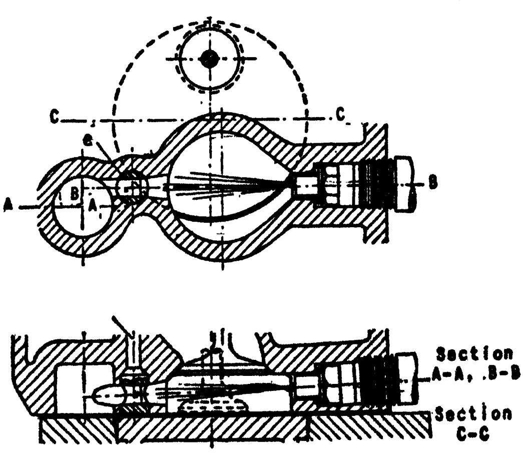

Fig. 11-14.—Combustion chamber with a Ramsey energy cell.

11-4. Combustion-chamber Characteristics.—The two main functions of a combustion chamber may be subdivided into the following detailed requirements:

1. An efficient preparation of the fuel-air charge which means:
   - (*a*) an even distribution of the injected fuel throughout the combustion space;
   - (*b*) a thorough mixing of the fuel with the air to ensure complete combustion with the minimum excess air possible.
2. An efficient smooth combustion, which depends upon:
   - (*c*) a sufficiently high temperature to effect ignition;
   - (*d*) a small delay period or ignition lag;
   - (*e*) a moderate rate of pressure rise during the second stage of combustion;
   - (*f*) a controlled, even burning during the third stage;
   - (*g*) a minimum of afterburning;
   - (*h*) minimum heat losses to the walls.

Conditions (*a*) means a directed flow or swirl of the air; condition (*b*) means an air swirl or squish of high intensity. Condition (*c*) is controlled by the selection of the proper compression ratio. Condition (*d*) was discussed in detail in Sec. 10-3; since one of the most important requirements is a high temperature of the charge, a hot spot or a hot insert insulated from the rest of the cylinder head, as shown in Fig. 11-11, *a* or *c*, or 11-13*b*, may be used to advantage. As mentioned before, a moderate rate of pressure rise during the second stage of combustion, (*e*), is a characteristic of a given combustion chamber rather than a condition which can be obtained by special means. Condition (*f*) is governed by the rate of injection, *i.e.*, by the injection system in conjunction with conditions (*a*) and (*b*); so is condition (*g*). Finally, condition (*h*) can be fulfilled by reducing the surface-volume ratio and also by decreasing the air swirl, which, however, contradicts condition (*b*).

The main characteristics of an injection system that link it with a given combustion chamber are atomization, penetration, fuel distribution, and the shape of the fuel spray.

11-5. Comparison of Chambers.—How well a certain combustion chamber fulfills the above listed requirements can be determined only from actual performance of the engine.

The general experience may be summed up as follows:

Precombustion chambers can be used for relatively high speeds, do not require high injection pressure, have a relatively moderate rate of pressure rise, but give a low mep and high fuel consumption.

Open chambers with a constant-pressure injection are suitable for more moderate speeds; they have a high rate of pressure rise, require high injection pressures, and have a multihole fuel nozzle which is easily clogged up.

The use of individual pumps increases the maximum speed limit but does not appreciably change the rest of the performance, except to give higher combustion pressures.

Two-stroke engines, owing to the scavenge turbulence, permit moderately high rotative speeds, but require high compression ratios and, at least with one type of unit-injectors, have very high injection pressures.

Subdivided chambers permit the use of high speeds, have a moderate rate of pressure rise, permit the use of single-hole nozzles and are satisfactory for small and medium-size engines of all speed ranges.

Air-cell chambers with a certain injection system, Fig. 11-12*b*, give only moderately high mep but are satisfactory for small cylinder sizes.

Energy cells are suitable for small and medium-size high-speed engines and, when properly designed, give an all-around satisfactory performance. However, the design of an efficient energy cell for an engine of a given bore, stroke, and speed, especially if the engine has to operate with variable speeds and loads, requires considerable experimentation and changes before the best size and shape can be found.

Conclusion.—In general the present engines indicate a definite trend toward higher speeds and with it a trend away from the quiescent combustion chambers toward the turbulent ones, either in the form of a subdivided chamber or an energy cell. At the same time, advantage must be taken of the various types of suction turbulences, especially since their intensity increases with the engine speed.

11-6. Problems.—1. A 11-in. × 18-in. Rathbun-Jones engine with a Price combustion chamber, Fig. 11-4, runs at 400 rpm; the mechanical clearance is ⅛ in.; the total compression ratio is 14.5:1; the throat is a circlof $3\frac{1}{4}$ in. in diameter; the connecting rod is 40.5 in. long. Compute the mean throat velocities of every 5° of crank travel from 45° before and up to the dead center. Assume $t_{ch} = 110$ F. The procedure of Table 11-1 or a simpler one mav be used.

2. A 16-in. × 20-in. De La Vergne type VM engine, Fig. 11-11*b*, runs at 360 rpm; the mechanical clearance is $\frac{1}{8}$ in.; the total compression ratio is 14.0:1; the throat has an approximately rectangular shape $3\frac{3}{8} \times 4$ in.; the connecting rod is 47.5 in. long. Lay out a curve of throat velocities from 45° before dead center for every 5° up to the dead center. Use a procedure similar to that of Table 11-1, and also try a simpler one based on volumetric relations alone, for comparison.

3. A $3\frac{3}{4}$-in. × 5-in. engine with a Ricardo turbulent combustion chamber, Fig. 11-11*a*, runs at 2200 rpm and has a mechanical piston clearance of $\frac{3}{32}$ in.; the total compression ratio is 16:1; the volume of the scoop in the piston is 17 per cent of the total compression space; the rectangular throat is $\frac{3}{8}$ in. wide by $\frac{7}{8}$ in. long; the connecting rod is 10 in. long. Lay out a curve of throat velocities from 60° before dead center for every 10° up to the dead center.

4. A 4-in. × $4\frac{1}{2}$-in. engine with a Hercules combustion chamber, Fig. 11-6, runs at 2600 rpm and has a mechanical piston clearance of $\frac{5}{64}$ in.; the total compression ratio is 16.2:1; the rectangular connection between the cylinder and combustion chamber is $1\frac{1}{4}$ in. wide and $\frac{11}{16}$ in. high; the connecting rod is 9 in. long. Lay out a curve of throat velocities similar to Fig. 11-6 from 45° before up to the top center.

5. A $4\frac{1}{2}$-in. × $5\frac{1}{2}$-in. engine with a recess in the piston, Fig. 11-7*a*, runs at 1800 rpm.; the total compression ratio is 16.25:1; the mechanical clearance is $\frac{1}{16}$ in.; the diameter of the cylindrical recess is $2\frac{7}{16}$ in. At the beginning of the compression the temperature and pressure of the charge are 135 F and 13.25 psia, respectively; the connecting rod is $11\frac{3}{4}$ in. long. Using the procedure similar to that given in Table 11-1, compute: (*a*) the air squish velocities for the last 35° before dead center and (*b*) the corresponding pressure heads; present these values in the shape of curves.

6. Work problem 5, assuming that the clearance between the cylinder head and piston at dead center is reduced to $\frac{3}{64}$ in., with a corresponding change in the compression ratio.

7. Work problem 5, assuming that mechanical clearance at dead center is reduced to $\frac{1}{32}$ in., with a corresponding change of the compression ratio.

8. Work problem 5, assuming that the top of the piston is changed and the diameter of the recess is increased to 3 in., all other data remaining the same.

9. A 12-in. × 16-in. engine with an open combustion chamber, Fig. 11-9*c*, runs at 327 rpm and has a clearance of $\frac{1}{8}$ in. between the rim of the piston and the cylinder head; the compression ratio is 13.0:1; the radial thickness of the piston rim is $1\frac{1}{4}$ in.; the connecting rod is 36 in. long. The pressure and temperature of the charge at the beginning of the compression stroke are 13.5 psi and 138 F, respectively. Using a method similar to the one given in Table 11-1, compute: (a) the air squish velocities for the last 45° of crank travel and (b) the corresponding pressure heads; present the results in the shape of curves.

10. Work problem 9, assuming that the mechanical piston clearance is reduced from $\frac{1}{8}$ to $\frac{1}{16}$ in., with a corresponding change of the compression ratio, all other data remaining the same.

11. A $4\frac{1}{2}$-in. × $5\frac{1}{4}$-in. engine with an air cell, Fig. 11-12*a*, runs at 1000 rpm; the mechanical clearance is $\frac{1}{16}$ in.; the compression ratio is 16.0:1; the volume of the air cell is 30 per cent of the total compression space; the diameter of the opening in the cell is $\frac{9}{32}$ in.; the connecting rod is $11\frac{1}{16}$ in. long. Assume that when the crank is 15° past dead center the pressures both in the main chamber and in the air cell are equal to 920 psig; the pressures at 20, 25, 30, 35, and 40° correspond to instantaneous exponents $n = 0.4, 0.8, 0.95, 1.09$, and 1.22, respectively. Compute (*a*) the turbulence velocities created by the receding piston and (*b*) the approximate pressure heads in the air cell.

12. Work problem 11, assuming that the opening in the cell is reduced to $\frac{1}{4}$ in., its volume to 25 per cent, and the speed is raised to 1100 rpm, all other data remaining the same.

13. Work problem 11, assuming that the opening in the cell is made only $\frac{7}{32}$ in. in diameter and the cell volume is reduced to 20 per cent of the total compression space; the engine speed is 1200 rpm, and all other data remain the same.

14. A $4\frac{7}{8}$-in. × 6-in. engine with an air cell, Fig. 11-12*b*, runs at 1800 rpm.; the mechanical clearance is $\frac{3}{64}$ in.; the compression ratio is 16.75; the volume of the air cell is 10 per cent of the total compression space; the opening in the cell is $\frac{3}{32}$ in. Assume that when the crank is 15° past dead center the pressures both in the main chamber and in the air cell are equal to 875 psig; the pressures at 20, 25, 30, 35, and 45° correspond to instantaneous exponents $n = 0.4, 0.8, 0.95, 1.10$, and 1.23, respectively. Compute (*a*) the turbulence velocities created by the air cell when the piston recedes and (*b*) the approximate pressure differences in the air cell.

15. Work problem 14, assuming that the engine speed is lowered to 1000 rpm, and the mechanical clearance is reduced to $\frac{1}{32}$ in.; the maximum pressure, due to the increased compression, is raised to 910 psig, and all other data and the dimensions of the air cell are unchanged.

CHAPTER 12

ENGINE PERFORMANCE

12-1. Factors Affecting Performance.—The word *performance* when referring to an engine, has several meanings and therefore when used should be clearly defined. In some cases it is used to designate the relationship between power, speed, and fuel consumption. In other instances, it may refer to the relationship between power, speed, and smoothness, or noiselessness, of operation. In the present chapter the word performance is used to designate the influence upon the indicated power developed by actual engines of those factors by which they differ from ideal engines.

The working process in an actual engine differs from a theoretical cycle in many respects: The working medium is not air but a mixture of air and fuel—gas, vapor, or finely atomized liquid combustion not only adds heat to the working medium but changes its chemical composition as well; specific heats of gases of the working medium vary considerably with temperatures; the theoretically constant-volume combustion is not instantaneous; there is no constant-pressure combustion in the combination-cycle engines; compression and expansion are not adiabatic, and heat is exchanged in both directions between the gases and engine-cylinder walls; the residual gases change the composition, temperature, and actual amount of the fresh charge. The latter is affected by several other factors: resistances in the intake and exhaust systems, engine speed, barometric pressure, and temperature and humidity of the intake air and fuel.

Most of these factors tend to decrease the thermal efficiency and power output of the actual engines. On the other hand, when taken into account, they show that the thermal efficiencies of the actual engines are not so much below the theoretically possible as it may seem at first.

12-2. Valve Resistance.—Applying theoretical cycles to four-stroke engines it is assumed that the exhaust and intake pressures are equal to the atmospheric pressure and that the exhaust and intake valves are opened and closed on dead centers. In actual engines the exhaust pressure is higher and the suction pressure is lower than atmospheric pressure on account of the resistances in the exhaust and intake pipes, ports, and valves. In order to reduce the back pressure during the exhaust stroke the exhaust valve opens several degrees of crank travel before the corresponding dead center, from 20 to 35° in slow-speed engines and up to 50° and even 65° in high-speed engines. The valves do not close instantaneously and therefore, in order not to throttle the exhaust too much when the exhaust valve begins to close, the full closing is effected a few, about 5 to 25, degrees after the other dead center, allowing for the total duration of the exhaust period 210 to 250° and even more, instead of the theoretical 180°.

The inlet valve is opened 5 to 10°, sometimes up to 35° before, seldom on, or a few degrees after dead center. The closing of the intake valve occurs after the next dead center and depends upon the rotative speed of the engine, 20 to 30° being used in slow-speed and up to 50 to 60° in high-speed engines, the duration of opening being about the same as for the exhaust.

When the piston during the exhaust stroke pushes out the combustion products, the pressure in the cylinder is p_c, Fig. 12-1, and the pressure difference $p_c - p_a$, where p_a is the atmospheric pressure, is necessary to overcome the exhaust-valve resistance.

When the piston begins the suction stroke, point c, first the residual gases in the compression space expand to a pressure p_d; after this the receding piston begins to draw the fresh charge into the cylinder. The pressure difference, or vacuum, $p_a - p_d$, is required to overcome the resistance in the intake valve and other parts of the intake system.

The increase of back pressure during the exhaust is less objectionable, so far as the engine power is concerned, than the increase of vacuum during the suction. If the average mean indicated pressure of an engine is 100 psi, then each 1 psi back pressure means a decrease of 1 per cent of the mean pressure and power of the engine. On the other hand, each 1 psi of drop below atmospheric pressure during suction decreases the weight of the fresh charge, and hence the power which can be developed, approximately by $(1/14.7) \times 100 = 6.8$ per cent.

In most engines the back pressure during exhaust is 0.5 to 1.0 psi; the pressure drop during suction in oil engines is also about 0.7 to 1.0 psi; in engines governed by means of throttling of the fresh charge the vacuum is greater—in low-speed gas engines the vacuum at full load varies from 1 to 2 psi; on high-speed carburetor engines at or near the maximum speed this vacuum may reach 4 and even 7 psi.

Volumetric efficiency is the ratio of the volume of fresh charge taken in during the suction stroke to the full piston displacement. The volume of the charge is taken at atmospheric pressure.

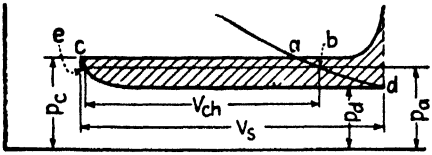

FIG. 12-1.—Suction diagram.

The volumetric efficiency η_v can be presented, Fig. 12-1, as

$$\eta_v = V_{ch}/V_s \tag{12-1}$$

The volumetric efficiency is affected by the resistance in the intake and exhaust valves and by the depression created by the throttle in engines governed by this method. The vacuum in the cylinders of a multicylinder automotive engine may run all the way from 3 in. Hg at full load and wide-open throttle to about 20 or 25 in. Hg when the engine is running idle.

The simplest way to determine η_v for an engine in operation is to take a weak-spring indicator diagram and to measure the lengths V_{ch} and V_s, Fig. 12-1. With a regular small-scale indicator diagram, the ordinates p_c and p_d can be measured more accurately than the length V_{ch}. In this case η_v can be computed, assuming with a sufficient accuracy that the exponents of the expansion curve $c - e$ and compression curve $d - a$ are equal to an average value $n = 1.35$. The volumes V_e and V_b can then be expressed by V_d and V_c, respectively, using equation (2-47).

Example 12-1.—Find the volumetric efficiency for a single-cylinder 7-in. × 8½-in. engine with a compression space of 72.7 cu in.; the exhaust back pressure is 1.3 psi, the intake vacuum is 1.2 psi. Assume an exponent of expansion and compression $n = 1.35$.

Using expression (12-1) and Fig. 12-1, where

$$V_s = 0.7854 \times 7^2 \times 8.5 = 327 \text{ cu in.}$$
$$p_c = 14.7 + 1.3 = 16.0 \text{ psia}$$
$$p_d = 14.7 - 1.2 = 13.5 \text{ psia}$$

by equation (2-47)

$$V_b = (327 + 72.7)(13.5/14.7)^{1/1.35} = 399.7 \times 0.9385 = 375.1 \text{ cu. in.,}$$
$$V_e = 72.7(16.0/14.7)^{1/1.35} = 72.7 \times 1.0644 = 77.4 \text{ cu. in.}$$
$$\eta_v = (375.1 - 77.4)/327 = 0.910$$

In low-speed high-compression oil engines with large valves, the volumetric efficiency η_v is around 0.92 to 0.95; in gas engines with a throttling governor, η_v is around 0.85 to 0.93, and in high-speed low-compression carburetor engines, it may be as low as 0.70 or even 0.60.

Pumping Losses.—During the exhaust and suction strokes the engine cylinder of a four-stroke engine acts as a pump—getting the exhaust products out and the fresh charge into the cylinder. The corresponding work represented by the cross hatched area on Fig. 12-1 is negative and is called *pumping losses.* When an indicator card is planimetered going over the lines of a card clockwise, the work of charging, the negative loop *ecd,* is automatically subtracted from the positive area and only seldom is the negative work actually determined. Depending upon the pressures p_c and p_d, which in turn depend upon the engine speed, the pumping losses decrease the available power in addition to the loss of power through a decrease of volumetric efficiency.

The pumping losses in four-stroke engines vary from 5 to 10 per cent of the indicated power. In two-stroke engines the power consumption of scavenge and charging pumps corresponds to pumping losses of four-stroke engines and varies from 7 to 12 per cent.

12-3. Residual Gases.—The fresh charge in an actual engine is diluted with exhaust gases left in the compression space from the previous cycle. These hot gases affect the fresh charge in two respects: (1) they lower the amount of fresh charge—both by lowering the volumetric efficiency of the suction stroke and by raising the temperature of the charge; and (2) they dilute the fresh charge, increase the amount of inert gases in it, and thus affect the ignition and combustion.

The temperature of the fresh charge t_{ch} after mixing with hot residual gases can be found equating the heat contents before mixing and after it

$$W_c c_p(T_c - T_{ch}) = W_{ch} c_p'(T_{ch} - T_a) \tag{12-2}$$

where W_c is the weight of the residual exhaust gases,
W_{ch} is the weight of the fresh charge,
T_c is the temperature of the exhaust gases, °R,
T_a is the temperature of the fresh charge before it enters the cylinder, °R,
c_p and c_p' are the specific heats of the corresponding gases.

The weights W_c and W_{ch} can be expressed from corresponding characteristic equations, with the designation of Fig. 12-1, as

$$W_c = 144 p_c V_c / R_2 T_c \tag{12-3}$$

and

$$W_{ch} = 144 p_a \eta_v V_s / R_1 T_a \tag{12-4}$$

where R_1 and R_2 are the gas constants of the fresh charge and products of combustion, respectively, and η_v is the volumetric efficiency. As a first approximation the change of specific heats before and after combustion may be neglected, or it may be assumed $c_p = c_p'$. Therefore, substituting expressions (12-3) and (12-4) in equation (12-2), solving it for T_{ch} and noticing that according to equation (4-38) $R_2 = R_1 n_2/n_1$, and $V_s/V_c = r - 1$, give

$$T_{ch} = \frac{p_c(n_1/n_2) + p_a \eta_v (r - 1)}{p_c(n_1/n_2)/T_c + p_a \eta_v (r - 1)/T_a} \tag{12-5}$$

and the temperature increase is

$$t_m = T_{ch} - T_a \tag{12-6}$$

Example 12-2.—Find the temperature increase of the charge due to the mixing with residual gases for the engine discussed in Example 12-1. The additional data are: fuel is gasoline, C_8H_{18}; 10 per cent excess air; temperature of residual exhaust gases $t_c = 1{,}120$ F; atmospheric temperature 70 F.

In order to use equation (12-5) the ratio of mols n_1/n_2 must be found:

Fresh Charge, Mols		Combustion Products, Mols	
C_8H_{18}	1	CO_2	8
$O_2 = (8 + {}^{18}\!/_4)1.1$	13.75	H_2O	9
$N_2 = 3.76 \times 13.75$	51.7	$O_2 = 13.75 - 12.5$	1.25
	$n_1 = 66.45$	N_2	51.7
			$n_2 = 69.95$

Thus $n_1/n_2 = 66.45/69.95 = 0.95$. From Example 12-1 $\eta_v = 0.908$; the compression ratio

$$r = (327 + 72.7)/72.7 = 5.5$$

By expression (12-5)

$$T_{ch} = \frac{16.0 \times 0.95 + 14.7 \times 0.908 \times (5.5 - 1)}{16.0 \times 0.95/1580 + 14.7 \times 0.908 \times (5.5 - 1)/530} = 613 \text{ R}$$

or an increase, due to the residual gases, of $t_m = 613 - 530 = 83$ F.

These calculations are not absolutely accurate, since the residual gases are giving up part of their heat to the cylinder walls. For this reason it is useless to take into account the change of specific heat.

As can be seen from equation (12-5), in high-compression oil engines, $r = 12$ to 17, the temperature increase t_m will be considerably less, only about 0.3 of the one found in Example 12-2.

12-4. Heat-loss Conditions.—Heat exchange between the gases and the walls, or more exactly the parts of the engine coming in contact with the gases, takes place in both directions. During the combustion, expansion, exhaust, and the later part of the compression, the gases have a higher temperature than the engine walls and give up heat to the walls. This heat is transferred to the cooling water or air and carried away by them. Part of it, however, is returned to the fresh charge during the suction stroke and the first part of the compression stroke when the temperature of the gases is lower than the average temperature of the walls. Figure 12-2 shows the temperatures at full load at different points of the cycle inside the cylinders of three four-stroke engines, the first a heavy-duty (aircraft) gasoline engine, the second a natural-gas, and the third an air-injection compression-ignition oil engine. In spite of the difference of cycles, the general character of all three diagrams is very similar. The temperatures in the gasoline engine operating without excess air run considerably higher.

Figure 12-3 shows the temperatures inside the cylinder of a two-stroke compression-ignition oil engine at different loads, characterized by the mean indicated pressures p_m as marked on the diagrams. The diagram of normal load, $p_m = 83$ psi, is drawn in full lines.

The above diagrams are typical for different types of engines. The mean temperatures t_m, shown on these diagrams, though

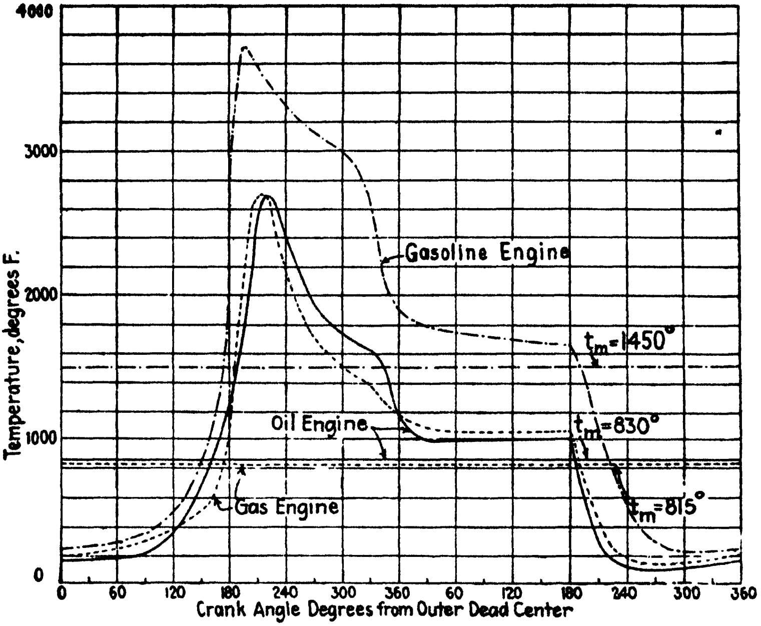

FIG. 12-2.—Temperature diagrams of four-stroke engines.

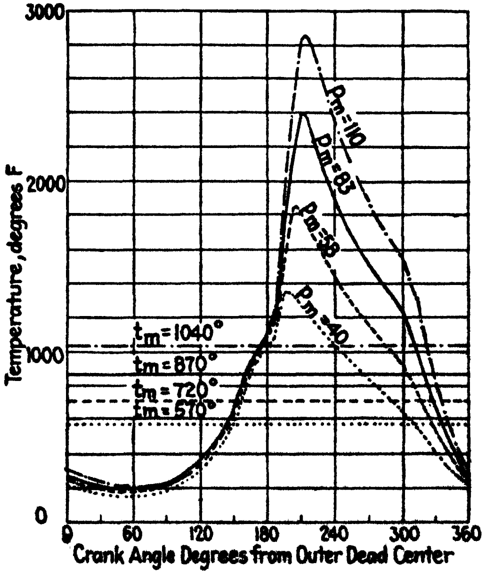

FIG. 12-3.—Temperature diagrams of a two-stroke engine.

slightly higher than the average temperatures t_w of the cylinder walls, give a good basis for considering the heat exchange between the working gases and cylinder walls.

Factors Affecting Heat Loss.—Heat transfer from the gases to the walls during combustion and expansion represents a loss which lowers the thermal efficiency of the engine. Heat loss during the exhaust stroke is unavoidable and represents the so-called *unavailable heat*—heat rejected to the refrigerator in any heat engine.

The heat lost to the walls during the second part of compression is practically equal to heat received by the gases from the walls during the first part of compression. These amounts can be neglected, since they are comparatively small.

The more important factors that affect the heat loss to cylinder walls in all engines are as follows:

1. Duration of combustion of the charge.
2. Temperature of combustion, which in turn depends upon the fuel, ratio of compression, and the engine load.
3. Speed of the engine.
4. Shape of the combustion space.
5. Size of the cylinder.
6. Timing of the ignition in spark-ignition engines or of fuel injection in compression-ignition engines.

The amount of heat lost to the walls increases with an increase of the duration of combustion and also with an increase of the combustion temperature. The latter increases with the engine load and with an increase of the compression ratio. An increase of the engine speed decreases the duration of combustion in seconds and thus should decrease the heat loss to the walls. Very often, however, an increase of speed creates additional turbulences favoring heat transfer to the wall; therefore, if the combustion duration is not over 50° of crank travel, an increase of speed decreases the heat loss only very little.

The shape of the combustion space has a great influence upon the heat loss. The latter decreases with a decrease of the ratio of surface to volume. However, other factors, particularly turbulence and flame propagation, also affect considerably the heat loss to the combustion-chamber walls.

The influence of the cylinder size is rather indefinite, because an increase of the cylinder size decreases the ratio of surface to volume but increases the flame travel and hence the combustion duration and usually is accompanied by a decrease in engine speed.

The influence of ignition timing can be established from the previous analysis; proper ignition timing gives a quicker combustion, less afterburning, and a smaller heat loss to the walls.

Heat Loss to Walls.—The amount of heat transferred to the walls may be computed from the general expression

$$Q = h_m A (t_g - t_w) \tau \tag{12-7}$$

where h_m is the mean heat-transfer coefficient, Btu/sq ft-°F-hr,
A is the area of the walls exposed, sq ft,
t_g is the average temperature of the gases, °F,
t_w is the average temperature of the walls, °F,
τ is the time, hr.

The heat exchange between the gases and metal walls is affected by radiation and convection combined with conduction. These factors together with the temperature difference determine the value of the coefficient h_m.

Designating the effect of radiation by h_r and of convection and conduction by h_c, also called the surface coefficient, gives

$$h_m = h_r + h_c \tag{12-8}$$

Application of the Stefan-Boltzmann law of radiation for these conditions gives

$$h_r = \frac{C}{t_g - t_w} \left[\left(\frac{T_g}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right] \tag{12-9}$$

where $T_g = t_g + 460°$ and $T_w = t_w + 460°$.

According to Nusselt's research work on gas engines,[1] the coefficient $C = 0.013$.

The surface coefficient h_c depends upon the movement of the gas, which is a function of piston speed c, in feet per minute, of the density of the gas which is a function of pressure p, in pounds per square inch, and its temperature t_g. According to Nusselt's investigations, the mean value for an engine cycle is[2]

$$h_c = b(160 + c) \sqrt[3]{p^2 T_g} \tag{12-10}$$

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[1] NUSSELT, WILHELM, Der Wärmeübergang in der Verbrennungskraftmaschine, *Forschungsarbeiten*, Heft 264, 1923.

[2] DUBBEL, H., "Oel und Gasmaschinen," p. 66, Julius Springer, Berlin, 1926.

where the coefficient $b = 0.0002$ in engines with low turbulence and may go up to 0.0004 in engines having a high turbulence.

Using expressions (12-8) to (12-10) in expression (12-7) gives

$$Q = C\left[\left(\frac{T_g}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4\right]A\tau + b(160+c)(\sqrt[3]{p^2T_g})A(t_g - t_w)\tau \quad (12\text{-}11)$$

By using expression (12-11) it is easy to show that the loss of heat by radiation is comparatively small, under normal conditions less than 10 per cent of the total loss. However, if the maximum temperature is increased above a certain limit, the thermal efficiency will be affected since the heat loss increases with the fourth power.

Other Factors.—Coefficient h_m is highest during combustion and in high-speed spark-ignition engines may reach 180 to 300 Btu/sq ft-°F-hr. In compression-ignition engines h_m is about 100 to 160 Btu/sq ft-°F-hr. During expansion the gas velocities are decreasing, and h_m drops to about 25 to 40. During the exhaust stroke the high gas velocities accompanied by turbulences in the exhaust ports raise h_m to 50 to 80. At low gas velocities, 2000 fpm, and low temperature differences, the value of h_m may drop to about 6. Sometimes, as a rough approximation, an average figure of 75 to 100 is taken for h_m over the whole period of heat loss, from the beginning of combustion to the end of exhaust.

If the total heat loss is to be found, then the area A in expression (12-7) must include the surfaces of the combustion space, the piston head, and with a sufficient accuracy one-half of the cylinder walls during the expansion and exhaust periods. If the loss through a certain surface is computed, A is taken for this surface alone.

The combustion-space surface depends upon the compression ratio and valve arrangement. In spark-ignition engines with r about 6, this area may vary from 1.8 to 3.0 piston areas; in high-compression oil engines it may shrink to about 1.3 of the piston area.

The mean temperature of the gases depends upon their temperatures at the beginning and end of the period and can be computed with a fair approximation, knowing the ratio of compression and composition of the charge. The mean temperatures of the walls depend naturally on the gas temperatures, heat-transfer coefficient, and cooling method. The following average figures may be used for spark-ignition engines, r about 5.5: combustion chamber 600 to 700 F; aluminum piston top 500 F; cast-iron piston 700 F; the inside surface of cylinder wall 50 to 80° higher than its outside surface; if the cylinder is water cooled, the average water temperature being 110 to 120 F, the inner cylinder surface is about 175 F. For compression-ignition oil engines, r about 14, average figures are: for cylinder head 500 to 600 F; cast-iron piston top uncooled 700 to 800 F; water or oil cooled 400 F; aluminum piston 450 F; cylinder liner about 500 F near the combustion space, dropping to about 150 F at the other end, or an average of 300 F in two-stroke engines and about 200 F in four-stroke engines.

The time τ may be conveniently expressed as the ratio of degrees of crank travel during the period for which the heat exchange is being computed to the crank travel of the complete cycle, 720° for a four-stroke and 360° for a two-stroke engine.

It would be wrong to charge against the engine the full amount of heat as found by means of expression (12-7) or (12-11). Only the available part of heat can be charged against the engine, the availability being found from diagrams Fig. 5-10 or 5-11, respectively, taking into consideration the average ratio of expansion during the particular period.

Example 12-3.—A four-stroke-cycle 7-in. × $8\frac{1}{2}$-in. gasoline engine runs at 840 rpm, develops 25 bhp per cylinder, operates on 110 per cent air, uses 0.5 lb/hp-hr gasoline of 19,480 Btu/lb heating value; compression ratio is 5.5; aluminum piston; gas temperature at end of compression $t_2 = 630$ F, after combustion $t_3 = 3700$ F, at end of expansion $t_4 = 2660$ F, during exhaust stroke $t_5 = 1500$ F. Determine the decrease of efficiency through heat loss to walls.

The duration of combustion, assuming a flame propagation velocity of 70 fps and travel across the cylinder is

$$\tau_1 = 7/(12 \times 70) = 0.00833 \text{ sec}$$

Duration of one revolution being $60/840 = 0.0714$ sec, τ_1 corresponds to $360 \times 0.00833/0.0714 = 42°$. Assuming that combustion starts 30° before dead center and that the exhaust valve opens at 40° before the lower dead center, duration of expansion becomes $180 - (42 - 30) - 40 = 128°$, leaving for the exhaust $180 + 40 = 220°$.

Area of piston top: $0.7854 \times 7^2/144 = 0.267$ sq ft
Area of combustion space: $0.267 + \pi \times 7 \times 8.5/[(5.5 - 1) \times 144] = 0.556$ sq ft
Area of side walls: $\pi \times 7 \times 8.5/144 = 1.297$ sq ft
Heat developed: $19{,}480 \times 0.5 \times 25 = 243{,}500$ Btu/hr

Heat loss during combustion: Assume h_m as 240; the temperature of the combustion space 700 F; piston top 500 F; heat loss, using expression (12-7).

$$\begin{aligned} Q_1 &= 240\{0.556[0.5(630+3700)-700]+0.267[0.5(630+3700)-500]\}42/720 \\ &= 17{,}600 \text{ Btu/hr, or 7.2 per cent of total heat} \end{aligned}$$

The data for combustion being the same as in Example Fig. 8-1, the average efficiency, as figured there, is $\eta_{id} = 0.343$, and the actual loss is

$$7.2 \times 0.343 = 2.5 \text{ per cent.}$$

Heat loss during expansion: Assume $h_m = 33$; temperature of cylinder side 175 F; exposed area of cylinder side can be determined taking in consideration movement of the piston, expression (8-3); however, a close enough approximation is to take the rounded up average between the two dead centers or $0.5 \times 1.297 = 0.649$ sq ft. Using the same expression (12-7),

$$\begin{aligned} Q_2 &= 33\{0.556[0.5(3700 + 2660) - 700] + 0.267[0.5(3700 + 2660) - 500] \\ &\qquad + 0.649[0.5(3700 + 2660) - 175]\} \times 128/720 \\ &= 23{,}700 \text{ Btu/hr, or 9.75 per cent of total heat} \end{aligned}$$

The average thermal efficiency can be taken also as $0.5(0.343 + 0) = 0.172$, giving for the actual loss 1.7 per cent.

Heat loss during exhaust: Assume $h_m = 60$, rest as above:

$$\begin{aligned} Q_3 &= 60[0.556(1500-700)+0.267(1500-500)+0.649(1500-175)]\times 220/720 \\ &= 28{,}800 \text{ Btu/hr or 11.8 per cent, all unavailable} \end{aligned}$$

The total heat transferred to the walls and from there to the cooling water is 70,100 Btu/hr or 28.8 per cent, which is a good average, indicating that our assumptions of temperatures and coefficients h_m were sufficiently accurate.

On the other hand, the actual loss is only $2.5 + 1.7 = 4.2$ per cent, the rest, or 24.6 per cent, being unavailable. Thus, deducting from the average ideal efficiency of combustion $\eta_{id} = 0.343$ the 4.2 per cent of available loss gives for the efficiency 30.1 per cent—very close to the indicated efficiency of $2545/(0.5 \times 19{,}480 \times 0.85) = 0.307$ found from above given fuel consumption and an assumed mechanical efficiency of 85 per cent.

12-5. Heat Flow from Walls.—The heat returned from the walls of a four-stroke engine to the fresh charge during the suction stroke raises the temperature of the charge, decreasing its weight and hence the power which the engine can develop. The amount of heat transferred can be computed from expression (12-7).

Because of a comparatively small gas density, the coefficient h_m is low, varying from about 6 to 30 Btu/sq ft-°F-hr, increasing with the engine speed. There exists an empirical expression for its computation, but the method is rather involved.[1]

The gas temperature t_g should take into consideration the temperature rise caused by mixing with the exhaust gases in the compression space, as discussed in Sec. 12-3.

The average temperatures of the different parts of the cylinder may be used as given before for the heat flow to the walls.

If t_d is the temperature and W the weight per hour of the mixture of the residual gases and of the fresh charge at the end of the suction stroke, then equating the amount of heat given up by the walls to the heat gained by the mixture gives

$$\Sigma h_m A(t_w - t_g)\tau = c_p W(t_d - t_{ch})$$

or

$$t_d - t_{ch} = \Sigma \tau h_m A(t_w - t_g)/c_p W \tag{12-12}$$

Taking advantage of the mol system, since it is easier to find volumes of gases than their weights, and substituting for $t_d - t_{ch}$ the designation t_h and for c_pW the sum of products of molecular heat by the number of mols, give for the temperature increase

$$t_h = \Sigma \tau h_m A(t_w - t_g)/(\gamma_{p1} n_{ch} + \gamma_{p2} n_c) \tag{12-13}$$

where n_{ch} and n_c may be found from equation (2-6), solving it for n

$$n = pV/1545T \tag{12-14}$$

Example 12-4.—Find the suction temperature t_d for the engine discussed in the Examples 12-1, 12-2, and 12-3.

The temperature in question can be found by means of expression (12-13). Coefficient h_m may be assumed as 25, since it is a high-speed engine; t_g is the average of the mixture temperature $t_{ch} = 153°$, as found in Example 12-2, and of t_d, which must be first estimated, and if the estimate was not close enough, then the computations must be repeated with a corrected t_d. Estimating $t_d = 183$ or $t_h = 30°$ gives for

$$t_g = 153 + 0.5 \times 30 = 168 \text{ F}$$

Time τ is one stroke out of four, or 0.25. Thus the numerator, heat given up by the walls,

$$Q = 0.25 \times 25[0.556(700 - 168) + 0.267(500 - 168) + 0.649(175 - 168)]$$
$$= 2432 \text{ Btu/hr}$$

The number of mols n_c of the residual gases per hour is by expression (12-14)

$$n_c = 16.0 \times 144 \times 72.7 \times 840 \times 60/(1728 \times 2 \times 1545 \times 1580) = 1.00 \text{ mol/hr}$$

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[1] STAMBULEANU, A., Wärmeübergang beim Ansaugehub von Verbrennungsmotoren, *Z. Ver. deut. Ing.*, vol. 81, No. 23, p. 670, 1937.

The number of mols ($n_{ch} + n_c$) of the fresh charge mixed with the residual gases, with the estimated final temperature $T_d = 613 + 30 = 643$ R, is

$$n_{ch} + n_c = 13.5 \times 144 \times 399.7 \times 840 \times 60/(1728 \times 2 \times 1545 \times 643) = 11.41 \text{ mol/hr}$$

this gives $n_{ch} = 11.41 - 1.00 = 10.41$ mol/hr.

The molecular specific heats for $T_g = 628$ R are found, using Table 2-1 and the composition of the gases from Example 12-2. For the fresh charge:

$\gamma_p(C_8H_{18}) \times 1 = 14.40 + 0.0533 \times 628 =$ 47.87
$\gamma_p(O_2) \times 13.75 = (11.515 - 172/628^{0.5} + 1530/628)13.75 =$ 97.3
$\gamma_p(N_2) \times 51.7 = (9.47 - 3.47 \times 10^3/628 + 1.16 \times 10^6/628^2)51.7.$ 356.1
$\Sigma(\gamma_p n) = 501.27$

$$\gamma_{p1} = \Sigma(\gamma_{p1}n)/n_1 = 501.27/66.45 = 7.54 \text{ Btu/mol-°R}$$

For the combustion products

$\gamma_p(CO_2) \times 8 = (16.2 - 6.53 \times 10^3/628 + 1.41 \times 10^6/628^2)8 =$ .. 74.8
$\gamma_p(H_2O) \times 9 = (19.86 - 597/628^{0.5} + 7500/628)9 =$ 71.7
$\gamma_p(O_2) \times 1.25 = (11.515 - 172/628^{0.5} + 1530/628)1.25 =$ 8.85
$\gamma_p(N_2) \times 51.7 =$ 356.1
$\Sigma(\gamma_p n) = 511.45$

$$\gamma_{p2} = \Sigma(\gamma_p n)/n_2 = 511.45/69.95 = 7.31 \text{ Btu/mol-°R}$$

Thus the denominator in equation (12-13) becomes

$$\gamma_{p1}n_{ch} + \gamma_{p2}n_c = 7.54 \times 10.41 + 7.31 \times 1.00 = 85.8 \text{ Btu/°F-hr}$$

and

$$t_h = 2432/85.8 = 28.3°$$

sufficiently close to our estimate of 30°.

The temperature of the mixture at the end of the suction stroke

$$t_d = 153 + 28.3 = 181.3$$

meaning a total increase of 111.3 F above the temperature of the charge before it enters the cylinder.

12-6. Charge Dilution.—Dilution of the fresh charge by residual gases is a function of the relative volume of the combustion chamber, of the valve resistance, particularly of the suction pressure, and of temperatures of the gases. Dilution is especially pronounced in engines controlled by throttling of the intake charge and in high-speed engines. Dilution is of particular interest in spark-ignition engines, whose ignition and combustion it affects. If the dilution exceeds a certain limit, the charge may begin to ignite irregularly and show considerable afterburning.

Numerically, dilution d_e is expressed as the ratio of the weight of the residual gases W_c to the total weight of the cylinder content, *i.e.*, residual gases W_c plus the fresh charge W_{ch},

$$d_e = W_c/(W_c + W_{ch}) \tag{12-15}$$

With the designation of Fig. 12-1

$$W_c = 144p_cV_c/R_cT_c \tag{12-16}$$

and

$$W_d = W_c + W_{ch} = 144p_d(V_s + V_c)/RT_d \tag{12-17}$$

Figure 12-4 illustrates the increase of the dilution with a decrease of the intake pressure p_d as it was found for a certain engine. In this diagram the weight $W_{ch} + W_c$ at 28 in. Hg is taken as 1.00, and all other weights are expressed as its fractions.

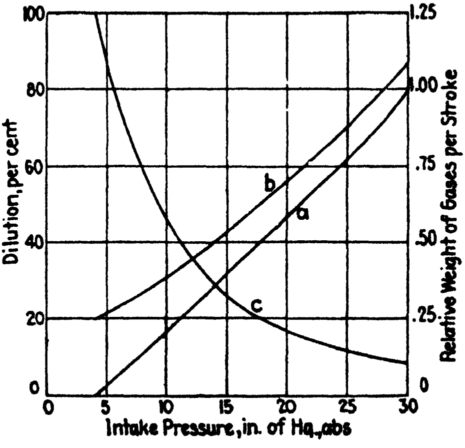

FIG. 12-4.—Intake pressure and charge dilution.

Curve *a*, Fig. 12-4, represents the fresh charge in the cylinder per stroke. The weight of the residual gases W_c increases with a decrease of the intake pressure, as the latter is the result of a decrease of the engine load and is accompanied by a lowering of the exhaust temperature, which is also the temperature of the residual gases. Adding to the ordinates of curve *a* the weights of residual gases gives curve *b*, which represents the total weight of the gases in the cylinder. Dividing the vertical distances between curves *a* and *b* by the ordinates of curve *b* gives curve *c*.

This curve shows the amount of dilution, which is quite considerable, especially at low intake pressures.

12-7. Charge Efficiency.—As already discussed, the resistances in the exhaust and intake valves reduce the amount of the fresh charge taken in. A further decrease of the charge is caused by its expansion due to mixing with the hot residual gases and to contact with the hot surfaces of the combustion chamber and cylinder.

The ratio of the weight W_{ch} of the actual fresh charge to the weight of the piston displacement V_s, if it were filled with a fresh charge having the outside pressure and temperature, is called *charge efficiency*[1]

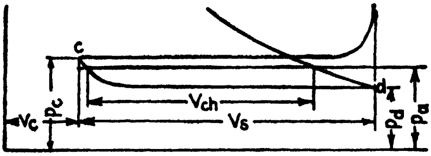

FIG. 12-5.—Suction diagram.

$$\eta_{ch} = W_{ch}/V_s w_a \tag{12-18}$$

The weight of the fresh charge W_{ch} can be expressed as the difference between the weights of the cylinder contents at the points d and c, $W_{ch} = W_d - W_c$, Fig. 12-5 and equations (12-16) and (12-17).

Noticing that the specific weight at outside conditions may be expressed as

$$w_a = p_a/RT_a \tag{12-19}$$

Substituting the corresponding values in expression (12-17) and simplifying it, with $R/R_c = n_1/n_2$, give finally

$$\eta_{ch} = T_a[rp_d/T_d - (p_c/T_c)(n_1/n_2)]/[p_a(r - 1)] \tag{12-20}$$

On account of the heat influences, η_{ch} is smaller than the volumetric efficiency η_v discussed in Sec. 12-2, and for the same reason the ratio η_{ch}/η_v decreases with an increase of the engine load. Charge efficiency η_{ch} is the characteristic, which is of value in analyzing the engine performance, its power output.

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[1] In American literature, charge efficiency is often called *volumetric efficiency*. This is confusing and not correct, as this efficiency deals with weights, not volumes of air.

However, for an engine in operation η_v is found more easily than η_{ch} and therefore may be used for estimating the latter.

Example 12-5.—Find the charge efficiency for the engine discussed in Examples 12-1, 12-2, and 12-4.

By equation (12-20)

$$\eta_{ch} = 530[5.5 \times 13.5/641.5) - (16.0/1580)0.95]/14.7 \times 4.5 = 0.784$$

The ratio $\eta_{ch}/\eta_v = 0.864$ gives an idea of the influence of the heating of the fresh charge during the suction stroke. The ratio η_{ch}/η_v may be higher or lower, see Table 13-1. The factors which influence η_{ch} will be discussed later.

The charge efficiency permits finding the amount of air available for combustion of the fuel in an engine of a given displacement and thus affects the mep of the engine. However, if the engine operates with an excess of air, like all compression-ignition oil engines, a certain amount of air after each cycle is left in the compression space, mixed with the residual gases; and the weight of actual available air is greater than found by expression (12-18). The more accurate expression is

$$\eta_{ch} = (W_{ch} + W_{ra})/V_s w_a \tag{12-21}$$

where W_{ra} is the weight of air in the residual gases and can be calculated from the exhaust-gas analysis in conjunction with other data used in expression (12-20).

12-8. Atmospheric Conditions. *Temperature. Pressure.*—The performance of internal-combustion engines is influenced by three atmospheric factors—temperature of the air, humidity of the air, and the barometric pressure. All these factors affect the air charge. According to the characteristic equation for gases, the weight of the air charge is proportional to the pressure p and should be inversely proportional to the absolute temperature T. However, taking into consideration the velocity in the intake port, the weight of the air charge will be found to be inversely proportional to $\sqrt{T}$, especially in high-speed automobile and aircraft engines. In accordance with this consideration, the expression[1] for correcting tests conducted under various conditions of barometric pressure and temperature is

$$N_c = N_0 \frac{p_s}{p_o} \sqrt{\frac{t_o + 460}{t_s + 460}} \tag{12-22}$$

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[1] Used by SAE, U. S. Bureau of Standards, and British Air Ministry.

where N_c is the corrected horsepower, N_o hp as found from test, p_o and t_o observed, and p_s and t_s are standard pressures and temperatures. Standard pressure $p_s = 29.92$ in. Hg; for t_s the SAE and U.S. Bureau of Standards give 60 F; the NACA rules give 59 F, and the ASME code 68 F.

The object of expression (12-22) is to give the horsepower which the tested engine would develop under standard conditions. Naturally expression (12-22) does not affect the thermal efficiency and N_c cannot be used instead of N_o in determining the fuel consumption per horsepower-hour.

For dealing with aircraft engines, the SAE Aircraft Engine Test Code modifies the equation (12-22) to

$$N_c = N_o \frac{29.92}{b + p} \sqrt{\frac{t_o + 460}{520}} \qquad (12\text{-}23)$$

where b is the dry air pressure, in. Hg, and
p is the air-scoop pressure, in. Hg.

In a great majority of cases $p_o < p_s$ and $t_o > t_s$, thus giving $N_c > N_o$. This increase is the result of additional weight of air under standard conditions. When an engine is working with a small excess of air or without any excess, as in the case of carburetor and some gas engines, a correction of the maximum horsepower is necessary. However, the correctness of using expression (12-22) for oil engines operating with a considerable excess of air, usually 40 per cent and more at sea level, is doubtful.

Humidity.—Theoretically the presence of water vapor in the fresh charge decreases the weight of air in a given volume of charge. This has been confirmed experimentally.[1,2]

Instead of computing the correction coefficient c as a ratio of volumes of humid air to dry air, c can be presented as a ratio of total pressure p_o to partial pressure of dry air $p_o - xp_w$, where p_w is the partial pressure of water vapor at observed temperature t_o, and x is the relative humidity

$$c = p_o/(p_o - xp_w) \qquad (12\text{-}24)$$

The influence of humidity is small, usually not over 2 to 3 per cent, and may be disregarded; however, it can easily be taken

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[1] GARDINER, A., *SAE Journal*, vol. 24, pp. 155, 239, 1929; vol. 25, p. 78 1929.

[2] BROOKS, D. B., *SAE Journal*, vol. 25, p. 277, 1929.

care of by combining expressions (12-22) and (12-24) into

$$N_c = N_o \frac{29.92}{p_o - xp_w} \sqrt{\frac{t_o + 460}{t_s + 460}} \qquad (12\text{-}25)$$

It may be mentioned also that an increase of the humidity of air used in carburetor engines requires a linear increase of the optimum spark advance. On engines of great excess of air the influence of humidity may be disregarded, the same as the influence of barometric pressure.

Expression (12-22) in conjunction with equation (7-11) given before allows the computation of the maximum horsepower N_o which an engine, rated N_c at sea level, can develop at an altitude of h ft if the same air-fuel ratio is used:

$$N_o = N_c \frac{p_o}{p_s} \sqrt{\frac{T_s}{T_o}} \qquad (12\text{-}26)$$

where $T_o = t + 460$ and t is found by expression (7-12).

The decrease of the air pressure with altitude has one beneficial effect on the performance of gasoline engines: The tendency to detonate decreases rapidly with the decrease of air pressure. Thus when the average octane requirement was about 68 at sea level, it decreased to 50 at 6000 ft altitude, and 18 at 12,100 ft.[1]

Compression-ignition oil engines usually operate with a considerable excess of air and therefore are less sensitive to a decrease of the intake-air pressure so far as maximum horsepower output is concerned. However, the rated horsepower at sea level usually represents the horsepower at which the fuel consumption per horsepower-hour is the lowest. This condition depends to a great extent upon the excess air present, and in order to maintain approximately the same excess air at higher altitudes, formula (12-23) may also be applied with a sufficient accuracy.

Two-stroke Engines.—While the operating conditions in these engines are more involved, in general a change of intake-air pressure, temperature, and humidity influence both the maximum power output and the fuel rate along the same lines as for four-stroke engines.[2]

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[1] McCOULL, HOLLISTER, and CRONE, Effect of Altitude on Antiknock Requirements of Cars, *Library Bull. of Abstracts*, Universal Oil Products Co., vol. 12, No. 47, Nov. 24, 1937.

[2] DOOLITTLE, J. S., Effect of Variations in Atmospheric Conditions on Diesel-engine Performance, *Trans. ASME*, vol. 63, No. 2, p. 94, February, 1941.

12-9. Entropy Diagrams. *Entropy Charts.*—The use of entropy for analyzing engine performance is made practical by means of the combustion entropy charts 4-2 to 4-9.

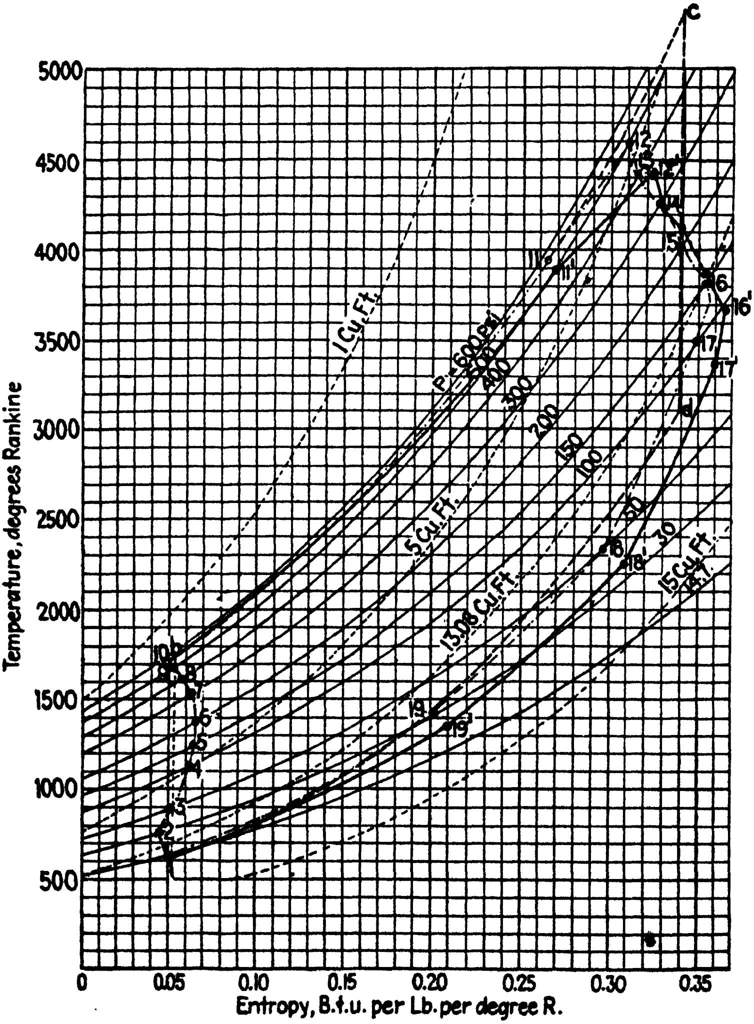

FIG. 12-6.—Temperature-entropy diagram.

The zero of entropy is arbitrary and can be located at any temperature level. In dealing with internal-combustion engines, this zero may be conveniently assumed at the standard temperature $T = 520$ R with the standard gas pressure $p = 14.7$ psia and a volume v corresponding to 1 lb gas at this p and T.

In Fig. 12-6 the center of the axes of the coordinates is taken at $T = 0$ R and $s = 0$. The scale of temperatures should be so selected as to have room for $T = 5500$ R; the scale of entropies should allow for $s = 0.45$ Btu/lb-°R or slightly more.

As already stated, adiabatic lines in a T-s diagram are straight, vertical lines; isothermal curves become horizontal lines.

Entropy Diagrams.—The use of an entropy chart may be illustrated on an indicator card of an air-injection oil engine, Fig. 12-7. First are drawn in the lines of absolute zero pressure and zero volume; next the card is crossed by a number of lines of constant pressure. After that the temperature at point 1, the beginning of the compression, is determined. With the known p_1 and T_1 the corresponding s_1 is found at their intersection on the corresponding entropy chart, in this case on Fig. 4-9, and the point 1 is located on Fig. 12-6. Point 2 is found by means of the characteristic equations for points 1 and 2, which give

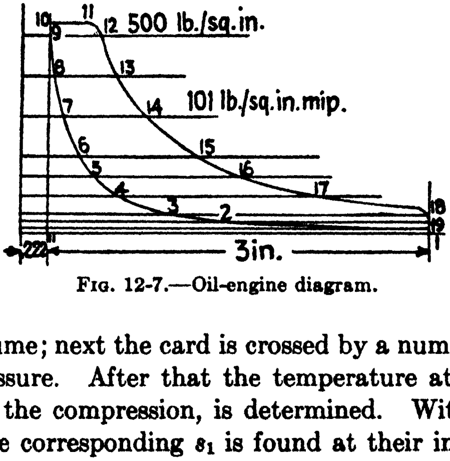

Fig. 12-7.—Oil-engine diagram.

$$T_2 = T_1 p_2 V_2 / (p_1 V_1) \tag{12-27}$$

Again, with T_2 and p_2 known, s_2 is found at their intersection on Fig. 4-9 and the point 2 is located on Fig. 12-6, and so on until point 10 is reached.

In finding points 11, 12 . . . 19, by means of characteristic equations, the temperatures are computed taking into account the change of gas constants R_1 to R_2, according to equation (4-38); and finally the addition to the weight of the charge W_{ch} of the weight of injection air W_a and fuel W_f is taken into account as shown by equation (12-28)

$$T_{12} = T_1(p_{12}V_{12}/p_1V_1)(n_1/n_2)W_{ch}/(W_{ch} + W_a + W_f) \qquad (12\text{-}28)$$

where n_1 and n_2 are the number of mols before and after combustion, respectively. The corresponding s is found from Fig. 4-9. The points thus located on Fig. 12-6 are connected, giving the contour 1-2-3 . . . 19-1. It should be remembered that, although the amount of heat rejected during the exhaust as presented on the T-s diagram is approximately correct, the exhaust line itself is fictitious in regard to the temperature. During this period the amount of exhaust gases in the cylinder changes, whereas in the diagram the amount is considered constant and the shrinkage of pressure and volume is presented as a result of change of temperature.

A good comparison is afforded by the theoretical T-s diagram 1-*b*-*c*-*d*-1 drawn for the same initial and working conditions as presented by the indicator card but without any heat losses.

12-10. Problems.—1. Find the volumetric efficiency for a 9-in. × 12-in. engine with a compression ratio $r = 5.75$; the exhaust back pressure is 2.5 in. Hg, and the intake vacuum is 2.3 in. Hg. Assume the exponent of the expansion of the residual gases $n_1 = 1.36$ and of the compression line $n_2 = 1.38$. Engine speed is 400 rpm, fuel is C_8H_{18}.

2. Find the volumetric efficiency for the engine of problem 1 if the intake vacuum is increased to 2.5 in. Hg.

3. Find the volumetric efficiency for a 6-in. × 7-in. engine with a compression ratio $r = 15.0$; the exhaust back pressure is 2.6 in. Hg, and the intake vacuum is 2.4 in. Hg; assume the exponents of the reexpansion and compression curves $n = 1.35$. Engine speed is 800 rpm, fuel is $C_{16}H_{30}$.

4. Find the volumetric efficiency for the engine of problem 3 if the compression ratio is raised to 17.0.

5. Find the temperature increase of the charge due to the mixing with residual gases for the engine of problem 1, using the following additional data: fuel octane; 5 per cent excess air; temperature of residual gases 1150 F; atmospheric temperature 80 F; barometer 29.4 in. Hg.

6. Work problem 5 using 15 per cent excess air and a temperature of residual gases of 1100 F; barometer 29.3 in. Hg.

7. Find the temperature increase of the charge due to the mixing with residual gases for the engine of problem 3, using the following additional data: fuel $C_{16}H_{30}$; 45 per cent excess air; temperature of residual gases 700 F; atmospheric conditions standard.

8. Work problem 7 with the compression ratio $r = 17.0$ given in problem 4 and 60 per cent excess air.

9. Find the approximate amount of heat transferred per hour per square foot by both radiation and convection from the gases to the uncooled piston top of a compression-ignition four-stroke engine; for the gas temperatures use those given in Fig. 12-2. Compare the heat flow: (*a*) over the whole cycle with (*b*) the one during the period when the gas temperatures are higher than their mean value. Assume a piston speed of 1000 fpm and mean indicated pressure of 100 psi.

10. Work problem 9, assuming that the piston is water cooled.

11. Find the approximate amount of heat transferred per hour by radiation and convection from the gases to the cylinder head of a two-stroke compression-ignition oil engine that has a bore of 28 in.; for the gas temperatures use Fig. 12-3; the mean temperature of the cylinder head surface can be assumed as 525 F at $p_m = 110$ psi, and 3 F lower for each 1 psi decrease of the mean indicated pressure. Investigate the two conditions (*a*) $p_m = 110$ psi and (*b*) $p_m = 58$ psi.

12. Work problem 11 for: (*a*) $p_m = 83$ psi and (*b*) $p_m = 40$ psi.

13. A four-stroke six-cylinder 3½-in. × 4⅛-in. engine develops 100 bhp at 3400 rpm; operating on 105 per cent air, it uses 0.51 lb/hp-hr gasoline; the compression ratio is 6.4:1; the engine has aluminum pistons; the gas temperature at the beginning of the compression is 185 F; at the end of combustion it is 3500 F; the temperatures at the end of compression, expansion, and during exhaust can be computed from above data with $n = 1.35$ and an exhaust back pressure of 1.1 psi. Determine the decrease of efficiency through heat loss to the walls.

14. A four-stroke eight-cylinder 3$\frac{1}{16}$-in. × 4¼-in. engine develops 117 bhp at 4000 rpm; operating on 110 per cent air it uses 0.505 lb/hp-hr gasoline; the compression ratio is 6.5:1; it has cast-iron pistons; the gas temperature at the beginning of the compression is 180 F; at the end of combustion it is 3550 F; the other temperatures can be computed with $n = 1.35$ and an exhaust pressure of 15.85 psia. Determine the decrease of efficiency through heat loss to the walls.

15. A four-stroke 9-in. × 12-in. oil engine with a compression ratio $r = 14.0$ develops 25 bhp per cylinder at 320 rpm; it has cast-iron, uncooled pistons; the gas temperature at the beginning of the compression stroke is 150 F; after the first combustion the pressure goes up to 600 psig; after the combustion is completed the temperature is 2800 F; the exponent of the compression stroke is $n_1 = 1.36$; of the expansion stroke it is $n_2 = 1.35$; the exhaust pressure is 15.9 psig. Assume the duration of the combustion from temperature relations and find the decrease of efficiency through the loss of heat to the walls during combustion and expansion.

16. A two-stroke 8½-in. × 10-in. compression-ignition oil engine with a compression ratio of 14.5:1 develops 60 bhp per cylinder at 600 rpm; it has aluminum pistons; the gas temperature at the beginning of the compression is 105 F; the pressure is 0.4 psi above atmospheric; after the first combustion the pressure goes up to 675 psig; at the end of the combustion the temperature is 3100 F; the exponent of the compression curve is $n_1 = 1.38$; of the expansion curve it is $n_2 = 1.35$. Assuming the duration of the combustion from temperature relations find the decrease of efficiency through the loss of the heat to the walls during combustion and expansion.

17. Find the suction temperature t_a for the engine and conditions as used in problems 1 and 5.

18. Find the suction temperature t_d for the engine and conditions as used in problems 1 and 6.

19. Find the suction temperature t_d for the engine and conditions as used in problems 3 and 7.

20. Find the suction temperature t_d for the engine and conditions as used in problems 3 and 8.

21. In an engine with a compression ratio 5.5:1 the pressure of the residual gases is 15.6 psia and their temperature is 1060 F when the suction pressure is 28.2 in. Hg; this temperature decreases 20° for every inch of intake pressure reduction. Find the dilution for the initial conditions and for intake pressures of 27.0 and 25.5 in. Hg.

22. Work problem 21 assuming that the compression ratio is raised to 6.25.

23. Determine the dilution for the compression-ignition oil engine discussed in problems 3 and 7.

24. Determine the dilution for the engine discussed in problems 3 and 8; the temperature of the residual gases is 675 F.

25. Find the charge efficiency for the engine discussed in problems 1 and 5 by using (*a*) equation (12-20) and (*b*) equation (12-21).

26. Find the charge efficiency for the engine discussed in problem 13, assuming that the suction vacuum is 1 psi.

27. Work problem 26 assuming that the suction vacuum is increased to 1.25 psi.

28. Find the charge efficiency for the engine discussed in problems 3 and 7; compare results by expressions (12-20) and (12-21).

29. Find the charge efficiency for the engine discussed in problems 3 and 24; compare results obtained by expressions (12-20) and (12-21).

30. During a test a gas engine developed 120 bhp; the barometric reading was 29.2 in., air temperature in the engine room was 74 F, and the relative humidity was 80 per cent. Find the horsepower which the engine will develop at standard conditions and in dry air.

31. During a test a compression-ignition oil engine developed 725 bhp; the power plant was located at an altitude of 6750 ft above sea level; the temperature in the engine room was 70 F and the relative humidity was 85 per cent. Find the horsepower which this engine would develop at standard conditions and with dry air operating with the same amount of excess air.

32. Find the power which a 200-hp natural-gas engine can develop at an altitude of 5200 ft, taking into consideration: (*a*) the pressure change alone and (*b*) pressure and temperature change.

33. Find the rated horsepower at sea level of an engine that must develop 450 bhp at an altitude of 6500 ft.

CHAPTER 13

FOUR-STROKE SPARK-IGNITION ENGINES

13-1. Suction.—In the following sections the performance of actual engines will be discussed chiefly concerning the behavior of the working gases, their pressures and temperatures, using information obtained from indicator cards taken from actual engines.

The events of the suction stroke are best investigated on an indicator diagram taken with a weak spring, Fig. 13-1. When the piston starts on its suction stroke, point *c*, the compression space V_c is filled with residual gases having a pressure p_c and temperature $T_c = T_e$. Their weight W_c can be found from the characteristic equation

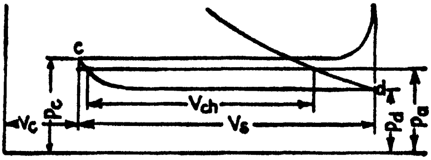

FIG. 13-1.—Suction diagram.

$$W_c = 144p_cV_c/R_eT_e \tag{13-1}$$

As the piston moves from *c* toward *d*, first the residual gases expand to the suction pressure p_d; after this, the receding piston begins to draw the new charge into the cylinder. The pressure difference $p_a - p_d$, where p_a is the atmospheric pressure, is required to overcome the resistance in the intake valve. This resistance increases with the engine speed or rather with an increase of the piston speed. While the suction line often is not a horizontal line, the real important factor is the pressure at the point *d*.

As already discussed, temperature t_d of the gases in the cylinder at the end of the suction stroke is higher than the outside temperature t_a. Mixing with hot residue gases raises it by t_m degrees and the contact with the hot cylinder and piston walls raises it by another t_h degrees, so that

$$t_d = t_a + t_m + t_h \tag{13-2}$$

These temperature increases can be computed from equations (12-5) for t_m and (12-12) for t_h.

The computation of the temperature t_d is necessary in order to find the charge efficiency η_{ch}, equation (12-20), which, in turn, is required in order to find the possible power output of the engine. However, the most important factor is the pressure at the point d, which cannot be computed but must be estimated on the basis of experience with engines having similar characteristics and operating conditions.

If sufficient data are not available for computing η_{ch}, some information may be obtained from the volumetric efficiency η_v, expression (12-1). On account of the heat influences already discussed, η_{ch} is always smaller than η_v, and for the same reason η_{ch}/η_v increases with a decrease of the engine load. The ratio η_{ch}/η_v varies from 0.95 for low-speed engines with a low compression ratio, down to about 0.76 for high-speed high-compression engines. Both η_{ch} and η_v usually are determined for full-load conditions for which they are of a particular, practical interest.

TABLE 13-1.—INTAKE DATA FOR SPARK-IGNITION ENGINES

Type of engine	Compression ratio r	Suction pressure p_d psia	Volumetric efficiency η_v	Charge efficiency η_{ch}	Ratio η_{ch}/η_v
Low speed..........	4.0–5.8	13.3–14.0	0.88–0.95	0.85–0.90	0.90–0.95
Medium speed......	4.0–5.8	12.4–13.4	0.84–0.91	0.75–0.85	0.85–0.94
Low speed..........	6.0–8.5	13.3–14.0	0.91–0.96	0.84–0.89	0.88–0.93
Medium speed......	6.0–8.5	12.4–13.5	0.87–0.92	0.72–0.84	0.82–0.91
High speed.........	6.0–8.5	11.5–13.0	0.83–0.89	0.60–0.79	0.76–0.89

Table 13-1 gives some average figures for p_d, η_v, and η_{ch} as found in actual engines. Under unfavorable conditions p_d may be considerably lower, which will result in correspondingly lower η_v and η_{ch}.

Figure 13-2 shows a sample indicator diagram from a slow-speed gas engine with a comparatively high compression, $r = 6$;

the volumetric efficiency $\eta_v = 0.93$. Figure 13-3 shows a diagram from an old type gasoline truck engine, $r = 4.0$, taken at about ¾ load, which accounts for the low volumetric efficiency $\eta_v = 0.69$. The dotted lines show conditions of full load, $\eta_v = 0.80$.

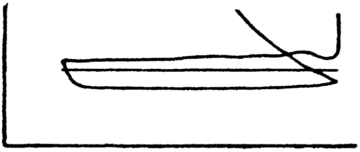

FIG. 13-2.—Diagram of a 20-hp gas engine, 180 rpm.

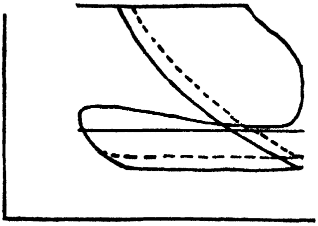

FIG. 13-3.—Diagram of a gasoline engine, 650 rpm.

13-2. Compression.—The pressure and volume at the beginning of the compression stroke are p_1 and V_1, respectively, Fig. 13-4, where $V_1 = V_s + V_2$, V_s is the piston displacement, and V_2 is the volume of the compression space. Remembering that the actual compression line is not an adiabatic but rather a polytropic curve, with the designation (5-1), the end pressure p_2 can be computed from the expression

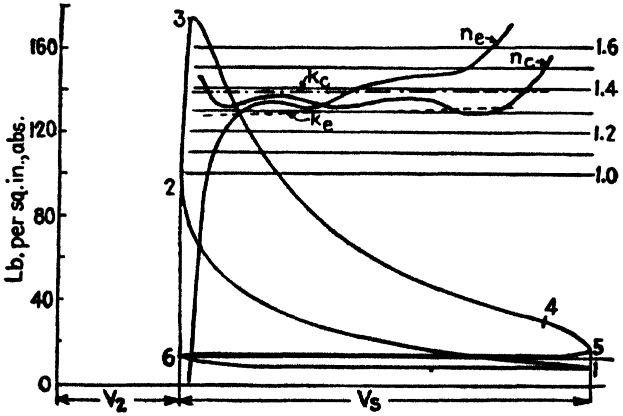

FIG. 13-4.—Gas-engine indicator diagram.

$$p_2 = p_1 r^n \tag{13-3}$$

and the end temperature, with $T_1 = T_d$,

$$T_2 = T_1 r^{n-1} \tag{13-4}$$

The heat exchange during the compression stroke is not uniform; hence the exponent n is not constant. During the first part of the compression the surface of the cylinder walls in contact with the charge is large, and a considerable part of the compression work transformed into heat goes through the cylinder walls into the cooling medium, and the temperature of the gases rises very slowly. Gradually, as the surface of the cylinder in contact with the gases decreases and the charge is confined by walls having a higher temperature, the heat exchange decreases. The temperature of the charge then begins to rise more rapidly.

Indicator diagrams taken from engines show accordingly that the first part of the compression curve has an exponent n lower than k; about the middle of compression, n approaches k but later may again begin to drop.

The n_c curve in Fig. 13-4 was obtained by the method of tangents as described in Sec. 2-5 and shows the variation of n during compression in this particular engine. The dotted line marked k_c represents the values of k at different points of the stroke, taking into consideration the changes of specific heats with the increase of temperature.

For most calculations, n can be assumed constant with a certain average value which depends upon the shape of the combustion chamber, engine speed, and ratio of compression r. The average values of n vary in different engines between 1.36 and 1.40, increasing with engine speed and decreasing with an increase of r. A fair average value for n is 1.35 for low-speed engines, and 1.38 for high-speed engines. In a case of poor cooling and high cylinder-wall temperature, n can even exceed 1.4, showing an increase of heat in the charge.

Leaks past the piston or through the valves lower the compression curve and decrease the apparent value of n computed from such a card. Excessive friction of the indicator piston distorts the compression curve in the same direction.

13-3. Combustion.—As already stated, in order to obtain combustion as close as possible to constant volume, ignition must occur before the piston reaches the dead center and must be accompanied by a sufficiently strong turbulence.

The temperature T_3 after combustion, point 3, is found from equation (2-61)

$$T_3 = T_2 + xQ/(c_vW) \qquad (13\text{-}5)$$

where x may be termed efficiency of heat generation and represents that part of theoretical heat which is actually imparted to the gases after deduction of all heat losses, primarily to the walls, and which is also due to incomplete combustion. These losses run from 7 to 12, sometimes up to 15 per cent, leaving for x values of 0.93 down to 0.88 and even to 0.85. The loss can be computed as shown in Example 12-3. The specific heat c_v may be determined as an average value for the temperatures T_2 and T_3. With compression ratios $r = 5$ to 7.5, T_3 may reach 4200 R.

The final pressure p_3 is found from the characteristic equations written for points 2 and 3,

$$p_2V_2 = WR_cT_2 \qquad (13\text{-}6)$$

and

$$p_3V_2 = WR_eT_3 \qquad (13\text{-}7)$$

where R_c is the gas constant of the charge before combustion and R_e of the gases after combustion, or of the exhaust gases. Dividing equation (13-7) by (13-6) and applying equation (4-38) gives

$$p_3 = p_2n_2T_3/(n_1T_2) \qquad (13\text{-}8)$$

If combustion is slow, as shown in Fig. 8-8, then the combustion line may be assumed to be a straight, inclined line 2-3, which is a polytropic line with an exponent n, which can be found from the indicator diagram. Instead of expression (13-5), the temperature T_3 at point 3 may be found from equation (2-57) as

$$T_3 = T_2 + xQ/(c_nW) \qquad (13\text{-}9)$$

where c_n is found from the expression (2-56).

The pressure p_3' at the end of a retarded combustion is computed as in the first case, designating by V_3 the volume at the end of combustion,

$$p_3' = p_2\frac{V_2n_2T_3}{V_3n_1T_2} \qquad (13\text{-}10)$$

On the other hand, for an existing engine, expression (13-5) allows finding the heat losses during combustion from

$$x = (T_3 - T_2)c_vW/Q \qquad (13\text{-}11)$$

where T_3 and T_2 can be calculated from the characteristic equations (13-6) and (13-7) for a given indicator card and Q can be computed from a fuel-consumption test.

13-4. Expansion.—Line 3-4, Fig. 13-4, also is not an adiabatic but rather a polytropic curve. As in the case of the compression line, the heat exchange during expansion is not constant. During the first part of expansion there is a considerable loss of heat to the water jacket which should result in an exponent n greater than k. However, in the majority of engines, combustion is not completed when the pressure begins to decline as the piston moves away from the firing dead center. Afterburning, especially noticeable when the turbulence is not sufficient, retards the temperature decline, decreasing the value of n during this part of expansion sometimes down to 1 and even below it. During the later part of expansion, n gradually increases and the average value for the whole curve usually varies between 1.30 and 1.39 with a fair mean value of 1.34.

Gas leaks past the piston, and the valves increase the apparent value of the exponent n; excessive friction in the indicator piston distorts the diagram in the opposite direction.

Curves n_e and k_e in Fig. 13-4, obtained by the same method as for the compression curve, illustrate the variation of n and its deviation from k_e.

The temperature T_4 at the end of expansion can be found from

$$T_4 = T_3(V_3/V_4)^{n-1} \tag{13-12}$$

where V_4/V_3 is only slightly less than r.

13-5. Exhaust begins before the piston reaches the dead center in order to obtain a smaller back pressure at the beginning of the return movement of the piston. The release, point 4, Fig. 13-4, occurs about 10 to 20 per cent and, in some aircraft engines with superchargers, up to 35 per cent of the piston stroke before dead center. Generally, the release is made earlier with the increase of the rotative speed of the engine.

The event of exhaust of the burned gases can be divided into two periods. During the first period, the gases escape into the exhaust line under the influence of the difference of pressures in the cylinder and in the outside atmosphere. This period lasts from the point of release and until the piston has traveled back approximately the same amount of its stroke. During this period, from 50 to 80 per cent, by weight, of the gases escape from the cylinder. During the second period, which lasts the rest, or about 90 per cent of the exhaust stroke, the balance of exhaust gases is pushed out by the piston, which is moving toward the compression end of the cylinder.

The first period occurs near the dead center, and its investigation can be best conducted using an offset indicator card taken in the same manner as for the investigation of ignition, except that a weak spring is used. Diagrams taken from different engines all have the same general character, as illustrated by diagram Fig. 13-5 taken from a natural-gas engine: the release occurs at *a* and the first period lasts to point *b*; the second period extends from *b* to the end of the stroke. A clearer picture of the entire event can be obtained from a diagram plotted from an offset card with the abscissas proportional to time elements or angles of crank travel, Fig. 13-6. The curve *a-b* always has a double curvature—the first part is concave; the second, convex to the line of abscissas. During the period *a-b*, the velocity of the escaping gases is a function of two variables: the area of the exhaust-valve opening, which gradually increases, and the pressure difference, which decreases.

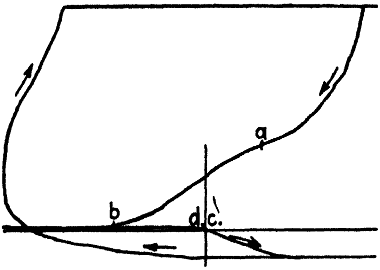

FIG. 13-5.—Offset weak-spring diagram.

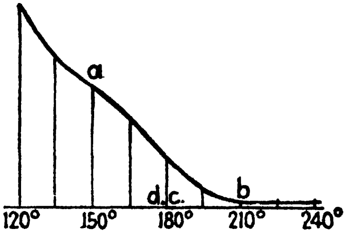

FIG. 13-6.—Exhaust-pressure curve.

The average gas velocity *v* during period *a-b* can be determined from the equation

$$K_e = Wv^2/2g \qquad (13\text{-}13)$$

where K_e is the kinetic energy of the gases that have escaped and W their weight. K_e or, rather, its heat equivalent, can be found from the heat balance of the engine. The weight W can be determined as the difference between the total weight W_a of the charge and the weight W_b found from the characteristic equation for point *b* where the pressure and volume are known, and the temperature T_b can be assumed to be equal to the temperature which the gases would have if they would be expanded to the volume V_b, Fig. 5-9, corresponding to the exhaust pressure at point b. This temperature can be either computed or found from the corresponding combustion chart.

From equation (13-13) with $g = 32.17$,

$$v = 8.02 \sqrt{K_e/W} \tag{13-14}$$

This velocity v determines the necessary dimensions of the exhaust valve.

During the second part of the exhaust, from b to c, the gas velocity v_2 is proportional to the piston velocity v_p and to the ratio of piston area A and exhaust-valve opening a,

$$v_2 = v_p A/a \tag{13-15}$$

The average back pressure p_e is from 5 to 10, sometimes 15, per cent higher than the outside atmospheric pressure. The increase of back pressure is undesirable because it increases the pumping loss and also raises the temperature of the cylinder walls, thus decreasing the charge efficiency during the suction stroke.

FIG. 13-7.—Weak-spring diagram.

Indicator diagrams taken with a weak spring sometimes show heavy pressure fluctuations, Fig. 13-7. The cause of such pressure waves usually is in the exhaust pipe line. If a long pipe line has no sharp turns, the inertia of the gas column put in motion by the impact of the high pressure and velocity at the moment of release can produce a temporary vacuum in the engine cylinder, sometimes a second vacuum later in the stroke. If the second vacuum comes near the dead center, it helps to scavenge the cylinder. However, such pressure waves change with the number of revolutions of the engine and cannot be depended upon to give material benefit to the engine performance.

The temperature t_e of the exhaust gases depends upon the air-fuel ratio and completeness of the combustion. Measured close to the exhaust port it may be as high as 1100 F with natural gas or gasoline as fuel and 400 to 500° higher with a rich mixture as used in aircraft engines.

13-6. Problems.—1. Determine the volumetric efficiency from the diagram of Fig. 13-1 and compare it with data of Table 13-1 to see whether or not it conforms to the latter; engine speed is 600 rpm.

2. A 10½-in. × 16-in. gas engine with a compression space of 346 cu in. runs at 220 rpm. The cylinder pressure at the beginning of the compression stroke is 0.75 psi below the atmospheric; the barometric pressure is 29.6 in. Hg; the atmospheric temperature is 73 F. Assume an average value, 1.36, for the exponent of the compression line; determine the pressure and temperature at the end of the compression.

3. The engine in problem 2 is governed by a throttling valve, and the vacuum in the cylinder at a certain load is increased to 1.25 psi; (*a*) find the pressure and temperature at the end of the compression under these conditions; (*b*) find the pressure and temperature if the vacuum is increased further to 1.75 psi.

4. If Fig. 13-4 presents the indicator diagram of a 10-in. × 14-in. gas engine, determine the pressure and temperature at the end of the compression; assume $n = 1.37$ and $t_a = 70$ F.

5. For the engine of problem 4 and Fig. 13-4 determine the temperature at the peak of pressure assuming that the molal ratio $n_2/n_1 = 0.95$.

6. For the engine of problems 4 and 5 find the temperature at point 4, Fig. 13-4, assuming an exponent of the expansion line $n = 1.36$.

7. For the engine of problem 2 (*a*) determine the average velocity of gases escaping during the first period, *a-b*, Fig. 13-5, of the exhaust; from test data, the pressure and temperature at point *b* were 15.7 psia, and 1100 F, respectively; point *b* corresponded to 8 per cent of the piston stroke from the dead center; the kinetic energy of the exhaust gases, found from the heat balance, was 4,000 Btu/hr; (*b*) find the percentages of gases that escaped during the period *a-b* and those expelled by the piston.

8. A six-cylinder 4-in. × 5-in. truck engine with a compression ratio $r = 5.75$ running at 2180 rpm gave, on the test stand, the following data: At a barometric pressure of 29.4 in. Hg, the weak-spring diagram showed a suction vacuum of 0.81 psi and an exhaust back pressure of 0.97 psi; the outside-air temperature was 69 F; the exponent of the compression $n = 1.37$, that of the expansion $n_2 = 1.35$; the maximum pressure $p_3 = 315$ psig; the point *b* of the exhaust curve was 10 per cent after the dead center; the kinetic energy of the exhaust gases, found from the heat balance, was 10,800 Btu/hr; (*a*) determine the average velocity of gases escaping during the first period of the exhaust; use the combustion chart for theoretical air; (*b*) find the percentages of exhaust gases that escaped and of those expelled by the piston. The exhaust temperature was 1100 F.

CHAPTER 14

FOUR-STROKE COMPRESSION-IGNITION ENGINES

Air-injection Engines

14-1. Intake.—What has been said about the intake of spark-ignition engines in general can also be applied to compression-ignition engines, with both air and airless injection.

Charge Efficiency.—According to equation (12-20), η_{ch} is influenced by the higher compression ratio r; however, a check will show that this influence is very small, as it lowers η_{ch} not over 1 per cent.

In compression-ignition engines, the induction system is shorter and presents less resistance than in carburetor engines.

The amount of heat picked up by the fresh air and the rise of its temperature are considerably smaller than in spark-ignition engines because of lower exhaust and cylinder-wall temperatures. On the other hand, the helpful shrinkage of the residual gases is also smaller. Thus these two factors practically neutralize each other.

The timing of the valves has the same effect as for spark-ignition engines, the normal closing of the inlet valve being about 20 to 30° after the dead center in low-speed and increasing up to 40° for high-speed engines.

In general the values for both η_{ch} and p_d are practically the same as given in Table 13-1.

14-2. Compression.—As repeatedly stated, the compression-ignition engines must have a high compression ratio in order to raise the temperature at the end of the compression stroke high enough to ignite the injected fuel.

This temperature depends upon the fuel used and also upon the pressure of the air into which the fuel is injected, decreasing with the increase of pressure. Figure 14-1 shows the relation between the pressures and temperatures of ignition for different oils. The characteristics of the fuel oil used in American oil engines are similar to the curves 1 and 2. Considering that an engine should be able to handle almost any oil which may be available and adding a safety margin of 100 F, the desirable temperatures seem to be not less than 850 F. To this must be added another 150 F to counteract the chilling effect of the expansion of high-pressure injection air in air-injection oil engines. To obtain a temperature of 1000 to 1100 F the compression pressure should be around 500 psi.

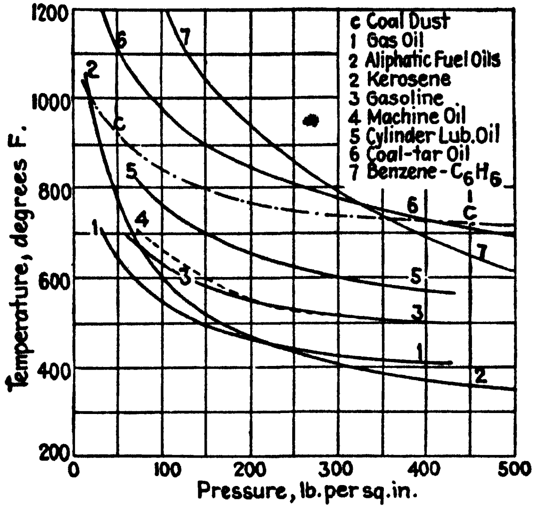

FIG. 14-1.—Effect of air pressure on ignition temperature.

The general character of the compression curve is similar to that in the explosion engines, with a greater decrease of the exponent n toward the end of the compression stroke.

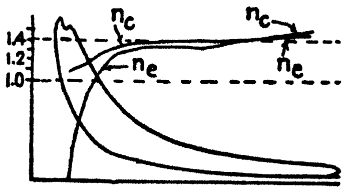

FIG. 14-2.—Oil engine—normal load.

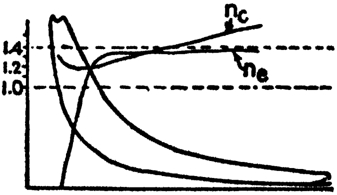

FIG. 14-3.—Oil engine—overload.

Figures 14-2 and 14-3 show samples of indicator diagrams of an oil engine with the n_e curves drawn in. Diagram Fig. 14-2 was taken at full load and Fig. 14-3 at 10 per cent overload. The higher temperatures of the cylinder walls in Fig. 14-3 influence the n_e curve and even have changed its general shape.

Engine speed, size of cylinder, and effectiveness of cooling influence the exponents of the compression line. Mean values of n vary from 1.35 for small, low-speed engines to about 1.39 for large, higher speed engines, and in most instances an average value of 1.37 may be used.

The compression pressures p_2, Fig. 14-4, may be computed as before

$$p_2 = p_1 r^n \qquad (14\text{-}1)$$

and the temperature

$$T_2 = T_1 r^{n-1} \qquad (14\text{-}2)$$

where at full load T_1 is 70 to 90 F higher than the room temperature. This is slightly lower than in spark-ignition engines, as a result of the smaller amount and lower temperature of the residual gases. The compression ratio r varies from 13 to 16.

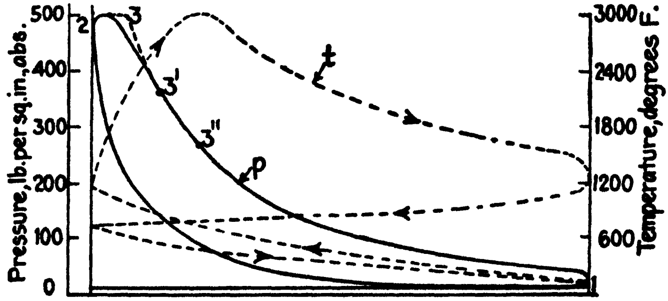

FIG. 14-4.—Pressures and temperatures in an air-injection oil engine.

14-3. Combustion.—Ignition does not start instantaneously, and a certain time element elapses before proper combustion begins. The fuel injection must occur a few degrees of crank travel before dead center in order to obtain an approximately constant-pressure combustion and to avoid a drop of pressure as the piston recedes. The amount of fuel-injection advance depends upon the characteristic of the fuel oil, the engine speed, method of injection, and some other features of the engine. It varies from 0 to 5° of crank travel in slow-speed air-injection engines and up to 9° in high-speed engines.

Ignition and combustion are best investigated by means of offset indicator diagrams. Figure 14-5 shows a combination card from an air-injection engine. Ignition is so smooth that it can be discovered, point a, only by increasing the fuel-valve lift and thus producing a slight rise of pressure, dotted line *c*. The dotted curve *d* shows expansion of air with the fuel shut off.

Figure 14-6 shows a diagram with late injection, due to insufficient fuel-needle lift.

Efficient turbulence, so necessary for good combustion, is created in these engines by the expansion of high-compressed injection air. This is one of the reasons that air-injection engines until recently had a lower fuel consumption than airless-injection engines.

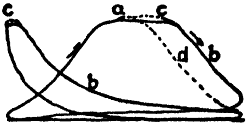

FIG. 14-5.—Regular and offset oil-engine diagrams

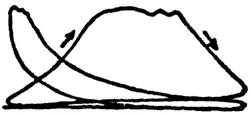

FIG. 14-6.—Late injection of fuel.

The temperature T_3 after combustion can be computed from equation

$$T_3 = T_2V_3/V_2 = T_2a \tag{14-3}$$

On the other hand, the same temperature can be computed from equation

$$T_3 = T_2 + xQ/c_pW \tag{14-4}$$

where all remarks made in regard to equation (13-5) apply again in full. Temperature T_3, computed from equation (14-4), is usually much higher than that from equation (14-3), which shows that combustion is not completed at point 3. Carefully conducted tests and analyses have proved that in practically all engines the pressure drop, point 3, begins before the end of fuel injection which occurs somewhere around point 3′, Fig. 14-4. Furthermore, combustion is never completed with the end of fuel injection and the afterburning extends farther into the expansion line.

Using the characteristic equation for different points of actual indicator cards, a curve of temperature can be plotted, and often the highest temperature is found at point 3″, considerably later than the end of fuel injection.

14-4. End of Cycle. *Expansion.*—The exponent *n* of the expansion curve is not constant—the afterburning at the beginning of the expansion causes it at first to be below 1, especially at full load and overload; toward the end of expansion, *n* increases up to 1.45 and even 1.50, Figs. 14-2 and 14-3. The average mean value of n for the whole expansion line is about 1.35.

The temperature t_4 at the end of expansion can be computed from

$$T_4 = T_3(V_3/V_4)^{n-1} \tag{14-5}$$

However, for an existing engine it is more certain to determine the temperature t_4 from the characteristic equation for point 4, taking pressure and volume from an indicator card and having determined by actual measurements the volume of the compression space and the weights of the air charge and fuel.

Exhaust.—All events occur along the same lines as in spark-ignition engines, with the difference that, regardless of a slightly higher maximum combustion temperature, the exhaust temperature is considerably lower. This is the result of a greater air-fuel ratio, greater expansion ratio, and termination of afterburning considerably earlier than in gas or gasoline engines in which, with a rich mixture, afterburning may extend even into the exhaust.

The back pressures encountered in these engines are the same as in spark-ignition engines.

Airless-injection Engines

14-5. Combination Cycle.—The present airless-injection compression-ignition oil engines operate on a cycle which theoretically is a combination cycle.

Intake.—All remarks made in respect to the intake stroke of spark-ignition and air-injection compression-ignition engines can be repeated here. The same applies to the charge efficiency, except when special turbulence-creating arrangements are used, such as the masking of the inlet valve mentioned in Sec. 11-2. In this case the charge efficiency is somewhat lowered. Figure 14-7 shows the influence of masking, for different lengths of the mask, 60, 90, and 120° of the port circumference, different position of the mask in respect to the inlet port, and speeds.

Compression.—The igniting of the fuel in airless-injection compression-ignition oil engines does not require such high temperatures as in air-injection engines, because of the absence of the chilling effect of the expanding injection air. Therefore, in moderate-speed engines the compression is carried only to 385 to 465 psia, in high-speed engines to 440 to 550 psia, which corresponds to compression ratios $r = 12$ to 14.5, and 14 to 17.

The exponent of the compression line is subject to the same remarks as in air-injection engines and depends to a great extent upon the engine speed. For open-combustion-chamber low-speed engines, a fair average may be assumed as $n = 1.35$, increasing it up to 1.38 for high-speed engines. Combustion chambers with heat-insulated surfaces give slightly higher exponents, up to about 1.38; and at high speeds and loads, 1.4 may be reached.[1]

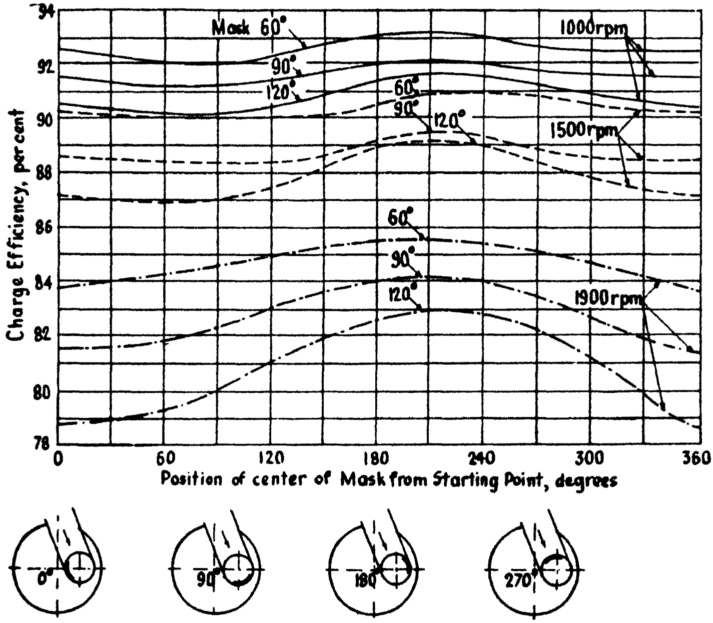

Fig. 14-7.—Effect of size and position of inlet-valve mask on charge efficiency.

Combustion.—Experience confirms the conclusion deduced from the theoretical analysis of the combination-combustion cycle—that the pressure rise during the constant-volume combustion raises the thermal efficiency. For this reason, as well as on account of the practical difficulty of obtaining a constant-pressure combustion with mechanical injection of the fuel, the combustion is always accompanied by a certain increase of pressure, giving a maximum pressure higher than in air-injection engines.

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[1] Dicksee, D. B., "The High-speed Compression-ignition Engine," p. 95, Interscience Publishers, Inc., New York, 1940.

Figure 14-8 shows a typical card from a mechanical-injection engine. Compression was carried to 390 psi, and burning raised it to about 560 psi. The offset card shows that ignition starts practically at dead center. However, as in air-injection engines, there must be a certain fuel advance *a*.

In low-speed airless-injection oil engines the angle of fuel advance is made from 6 to 10°. In high-speed airless-injection engines the angle of advance must be increased to 15 to 30°. This amount is necessary not only to allow for a slow-starting combustion but also to take care of a time lag between the moment of fuel-cam or fuel-distributor action and the moment when fuel actually enters the cylinder. This time lag is the result of lost motion, insufficient rigidity in the fuel-injection mechanism, resistance in the fuel pipes leading to the nozzles, and compressibility of the fuel oil. To secure uniform fuel admission in a multicylinder engine all fuel lines to the nozzles should be of equal length. The excessively long lines to nozzles located closer to the pump or distributor are bent in the form of loops or coils with large radii, to avoid additional hydraulic resistances.

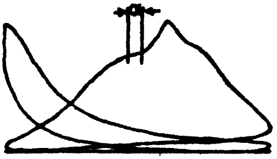

FIG. 14-8.—A mechanical-injection oil-engine diagram.

The combustion process was discussed in Chap. 10.

The rest of the cycle—expansion and exhaust—do not differ from those of air-injection engines.

14-6. Problems.—1. Compare the pressures and temperatures at the end of the compression for an engine with a compression ratio $r = 14$; the exponent $n = 1.36$; in one case, at the beginning of compression the vacuum is 0.7 psi and the temperature 80 F above outside temperature; in the other case, the vacuum is increased to 1.4 psi, with the same temperature. Assume standard barometric pressure and temperature.

2. Compare the pressures and temperatures at the end of the compression from problem 1 with those computed for compression ratios $r = 15.25$ and $r = 16.5$.

3. Present the results of problems 1 and 2 in the shape of curves.

4. Find the pressures and temperatures at the end of the compression for an engine with a compression ratio $r = 17$; the exponent $n = 1.37$, the temperature increase during the intake stroke is 76 F, the outside temperature is 72 F, and the barometer is 29.6 in. Hg; the intake vacuum is: (*a*) 0.85 psi and (*b*) 1.7 psi.

5. Work problem 4 assuming $n = 1.39$, the outside temperature 45 F and the barometer 29.8 in. Hg.

6. Determine the minimum compression ratio for an engine built to operate on fuel whose characteristic is curve 3-3, Fig. 14-1, using (*a*) air injection and (*b*) airless injection; assume an exponent of compression $n = 1.38$; outside air temperature 0 F.

7. Determine the temperature at the end of the first part of combustion in a combination-cycle engine with $r = 12.5$; the pressure rise $b = 1.5$; the exponent $n = 1.39$; the initial conditions are the same as in problem 1.

8. Determine the temperature at the end of the first part of combustion in a combination-cycle engine with a pressure rise $b = 1.4$; use the other data from problem 4.

CHAPTER 15

TWO-STROKE ENGINES

15-1. Types of Two-stroke Engines.—Two-stroke-cycle engines of different makes vary in two respects—the method of producing the scavenge air and the method of admitting it into the engine cylinder. However, practically all these engines have one feature in common, namely, that either the exhaust gases are discharged or the scavenge air is admitted, or both operations are performed through ports uncovered by the engine piston.

There are three different methods of producing scavenge air: (1) by an enclosed crankcase with the back of the engine piston used as a scavenge pump; (2) by a special built-in scavenge pump using the engine piston, either its back or a so-called *step piston;* (3) by a separate scavenge pump, either driven from the engine crankshaft or using outside power.

The first method is the simplest but also the least satisfactory, as the crankcase pump has a very low volumetric efficiency and, instead of an excess of air, the engine receives less than is theoretically necessary.

The second method is an improvement over the first one in respect to volumetric efficiency but has several drawbacks, chiefly in regard to the design of large engines.

The third method is the best and is being used in all larger and some small engines.

The different schemes for scavenging, meaning removing the burned gases and admitting the fresh charge, may be divided into two main groups: (*a*) return flow and (*b*) uniflow.

15-2. Return-flow Scavenging.—In this scheme the exhaust and scavenge ports are both opened and closed by the piston; the scavenge air is directed toward the other end of the cylinder by a slant of the scavenge ports or by the shape of the piston head or by both and then returns to the piston head pushing the burned gas before it. Actually some of the air mixes with the burned gases and escapes with them to the exhaust ports, thus lowering the scavenge efficiency.

Depending on the shape and relative position of the exhaust and scavenge ports, return-flow schemes may be subdivided into:

(*a*) cross-scavenging, Fig. 15-1*a*, the simplest but also the least efficient one;

(*b*) full-loop scavenging with exhaust ports above the scavenge ports, Fig. 15-1*b*;

(*c*) tangential-loop scavenging, Fig. 15-1*c*; and

(*d*) combination of loop- and cross-scavenging, Fig. 15-1*d*.

Loop scavenging is also called *backflow scavenging.*

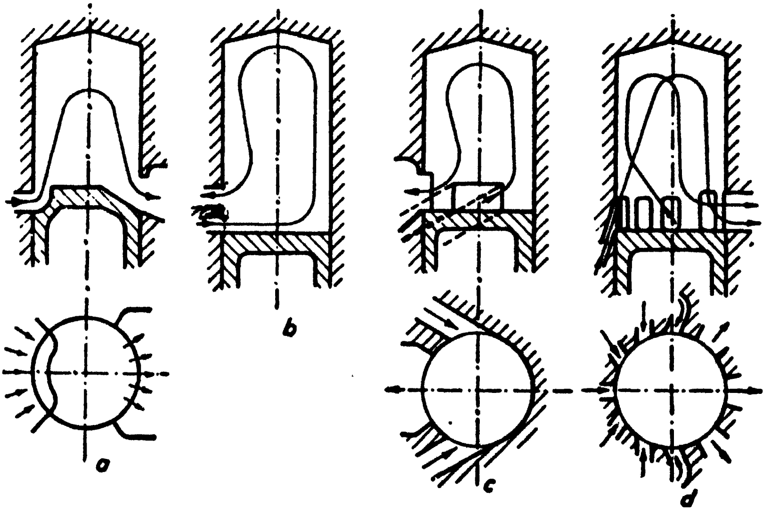

FIG. 15-1.—Schemes of return-flow scavenging.

Simple Port Scavenging.—Figure 15-2 shows the cylinder and ports of a cross-scavenged engine; *s* are scavenge ports and *e* exhaust ports. The timing diagram is shown in Fig. 15-3: *EO*, exhaust opens; *SO*, scavenge opens; *SC*, scavenge closes; and *EC*, exhaust closes. The same timing diagram applies for loop scavenging by simple ports.

This arrangement is very simple, especially when used in combination with crankcase scavenging. However, it has several disadvantages: The scavenge ports must be made shorter than the exhaust ports in order to permit the cylinder pressure to drop sufficiently before the piston begins to uncover the scavenge ports. On the return stroke the piston covers first the scavenge ports, Fig. 15-3, and the compression begins only after the piston has covered the exhaust ports. At this point the pressure in the cylinder is hardly any higher than in the exhaust line. The unavoidable mixing of scavenge air with the burned gases reduces further the fresh charge and results in a low mean effective pressure.

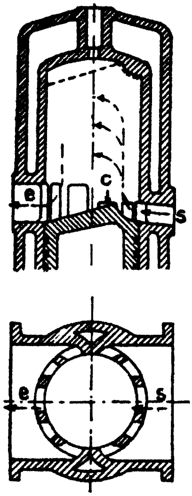

FIG. 15-2.—Sections of a simple port-scavenging engine.

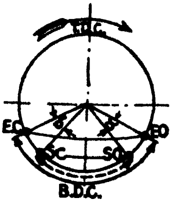

FIG. 15-3. Timing diagram of this engine.

Port Scavenging with Check Valve.—Large engines, for example, those built by Nordberg Manufacturing Company, use an improved cross-scavenging scheme, Fig. 15-4, originated by Sulzer Bros.: the scavenge ports are either of the same length or slightly longer than the exhaust ports. However, automatic check valves prevent the high-pressure exhaust gases from penetrating into the scavenge-air receiver. When the pressure inside the cylinder drops below the scavenge-air pressure, the latter opens the check valves, scavenging starts, and is continued until the ports are covered by the piston. The use of a separate scavenge-air pump permits having a desired excess air, thus

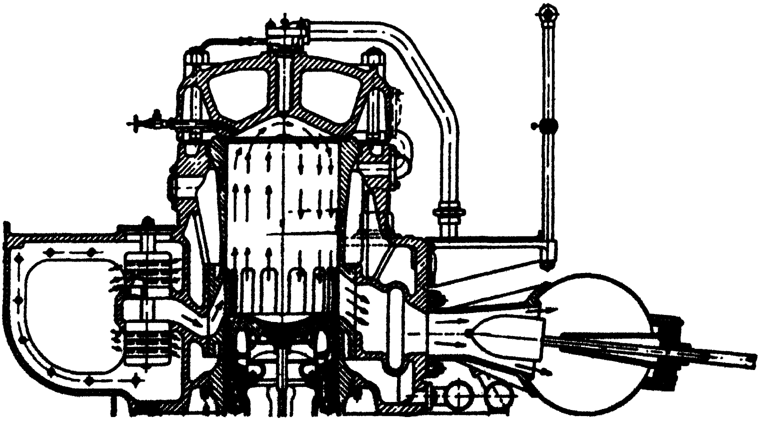

FIG. 15-4.—Cross-scavenging with check valve. (*Courtesy of Nordberg Manufacturing Company.*)

obtaining excellent scavenging, a high pressure at the beginning of the compression, and high mean effective pressures.

The timing diagram, Fig. 15-5, has this peculiarity: the moment when scavenging starts, *SO*, is not fixed but depends upon the pressure relation.

Combined Port and Valve Scavenging. Sulzer Scheme.—This scheme is a modification of the one with check valves. In this scheme the scavenge ports are divided lengthwise into two groups, *a* and *b*, Fig. 15-6. Ports *b* act in a way similar to scavenge ports of Fig. 15-1; ports *a* begin to admit air only after the mechanically operated scavenge valve *d* opens, point *SVO*, Fig. 15-7. On the upward piston stroke, scavenging is continued through ports *a* after the exhaust ports *e* are closed by the piston until the latter also covers ports *a*, point *SVC*. Valve *d* is closed after point *SVC* and before point *EO*.

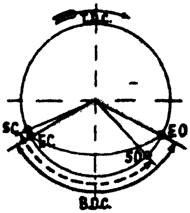

FIG. 15-5.—Timing diagram to Nordberg engine.

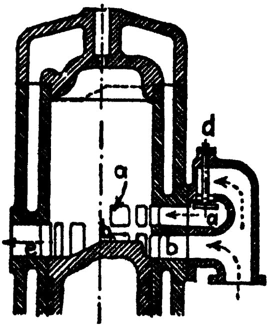

FIG. 15-6.—Combination valve and port scavenging.

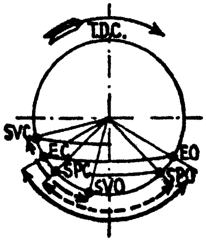

FIG. 15-7.—Timing diagram of this engine.

This scheme is successfully applied to medium- and large-size engines.

M. A. N. Scheme.—This is a loop-scavenging arrangement with oscillating valves in the exhaust lines and is used for large double-acting engines. When the piston uncovers the exhaust ports *e*, Fig. 15-8, the oscillating valve *u* is open; at the moment when, on its return stroke, the piston closes the scavenge ports *s*, valve *u* is also closed and is opened only when the piston again begins to uncover ports *e*. The piston length is equal to the stroke, the upper piston edge uncovering separate ports for the upper cylinder half. This arrangement allows the use of an early release of exhaust and at the same time prevents escape of scavenge air after closing of the scavenge ports. The design has proved very satisfactory for large engines operating with medium or high speeds. It gives good scavenging and high mean effective pressures. In this country it is used in H.O.R. diesel engines of large sizes, built by General Machinery Corporation, Hamilton, Ohio.

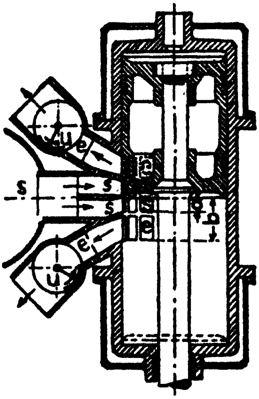

FIG. 15-8.—M.A.N. system of scavenging for a double-acting engine.

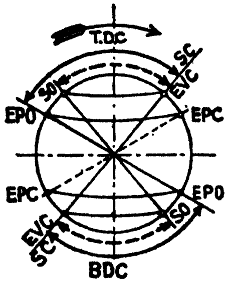

FIG. 15-9.—Timing diagram of a M.A.N. double-acting engine.

15-3. Uniflow Scavenging.—Exhausting the burned gases from one end of the cylinder and admitting the scavenge air from the other end gives a straight flow of the scavenge air. This decreases the tendency to form turbulences and reduces the mixing of the scavenge air with burned gases; thus the scavenge efficiency is increased and higher mean effective pressure can be obtained. The three main schemes of uniflow scavenging are:

(*a*) port and poppet-valve scavenging, one piston, Fig. 15-10*a*;

(*b*) port scavenging with opposed pistons, Fig. 5-10*b*;

(*c*) scavenging controlled by a sleeve valve, Fig. 15-10*c*.

An alternate scheme could be a combination of piston-controlled scavenge ports and a sleeve valve for the exhaust, or vice versa.

Port Scavenging with Opposed Pistons. Junkers Scheme.—Very effective scavenging and filling with a fresh charge are obtained in engines with opposed pistons, Fig. 15-11. The exhaust ports *a* are uncovered by the upper piston, the scavenge ports *b* by the lower one. All ports are distributed uniformly over the whole circumference. The tangential arrangement of the scavenge ports creates a spiral motion of the air charge which remains during the compression and contributes to a good combustion. The pistons move in opposite directions, being connected to a three-throw crankshaft—the lower piston to the center crank and the upper one through crossheads and con-

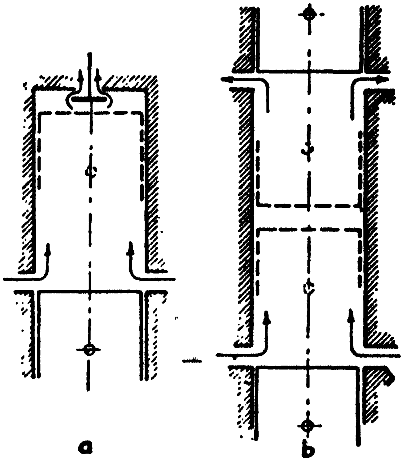

FIG. 15-10.—Schemes of uniflow scavenging.

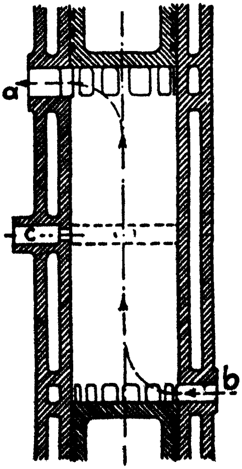

FIG. 15-11.—Junkers opposed-piston engine.

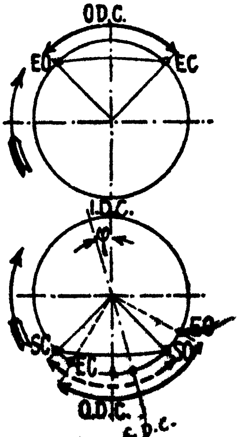

FIG. 15-12.—Timing diagram of this engine.

necting rods to the two parallel side cranks. In a later design the pistons are connected to two separate crankshafts and the upper crankshaft is connected to the lower one by spur gears.

In Fairbanks-Morse opposed-piston oil engines, the two crankshafts are connected by a vertical intermediate shaft and two pairs of bevel gears.

If the cranks of the two pistons form an angle of 180°, the scavenge-timing diagram is identical with Fig. 15-4. Slightly advancing the crank of the upper piston by an angle $\varphi = 10$ to 15°, the scavenge ports are made to close after the exhaust, and thus a supercharging effect is obtained, Fig. 15-12.

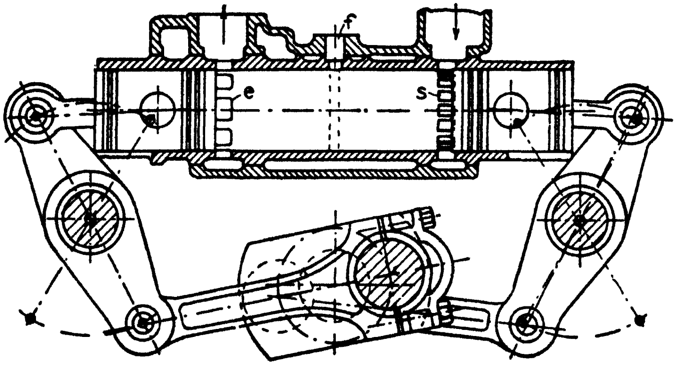

FIG. 15-13.—Opposed-piston engine with one crankshaft.

The advantages of this scheme are: very good scavenging and charging conditions, therefore high mean indicated pressures, simple and efficient combustion chamber, and good balancing of the reciprocating masses, even with one piston slightly leading the other.

However, these advantages are to a certain extent offset by mechanical disadvantages: complicated running gear, difficulties in cooling the pistons, and great height of the engine if the latter is of the vertical type.

This scheme is used only for medium- and large-size stationary, marine, and airplane engines.

One-crankshaft Scheme.—In this design, Fig. 15-13, both pistons act upon the cranks, located at 180° one to another, of the same shaft by means of rocker arms and connecting rods. The scheme lends itself best for horizontal engines. This design has another advantage: If the crankshaft is set slightly below or above the line connecting the lower ends of the rocker arms, the travel of one of the pistons near the dead center is retarded in respect to that of the other. With the arrangement of Fig. 15-14 the exhaust ports are uncovered by the left-hand piston 1 before the scavenge ports are uncovered by piston 2, and thus the necessary pressure release is obtained. On the return stroke, the exhaust ports are closed in advance of the scavenge ports, thus giving a certain degree of supercharging. The time element of the advance depends upon the angle α. The intrinsic balance of the reciprocating masses is also better than in the original Junkers scheme with one crank advanced in respect to the other one.

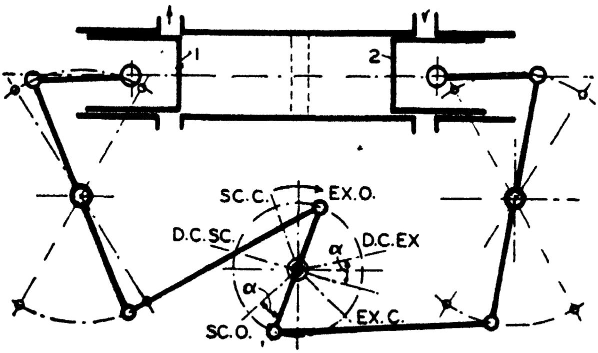

Fig. 15-14.—Lead of piston 1 in respect to piston 2 when approaching outer dead center.

Hill Diesel Engine Company is building engines according to this design here and Sulzer Bros. in Europe have developed the design independently.

One-piston Uniflow Scheme.—This scheme was originated many years ago with engines in which the burned gases were removed through ports controlled by the piston and scavenge air was admitted through poppet valves in the cylinder head. The turbulences created in the air when passing through the valves caused a mixing of the air with the burned gases, a poor scavenge efficiency, and relatively low mean effective pressures. A number of engines operating on this scheme are in use, and some are still being built, chiefly to run on gas fuel.

When, owing to metallurgical progress, exhaust valves were produced that stand up satisfactorily under the severe working conditions, high temperature, corrosive action of the burned gases, and impact when seating, a reversed scavenge scheme was tried, using poppet valves in the cylinder head for exhaust and scavenging through piston-controlled ports, Fig. 15-15. The results obtained were exceedingly satisfactory, and this scheme soon began to be applied to medium- and high-speed compression-ignition General Motors oil engines. The timing diagram, Fig. 15-16, is very flexible; the angles *a* of advance of release and *b* of closing the exhaust can be easily changed to suit speed and scavenge-air-pressure conditions. The scheme is particularly well adapted for supercharging.

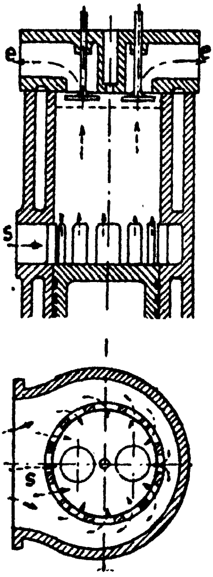

FIG. 15-15.—Sections of a uniflow engine with exhaust valves.

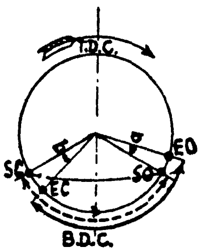

FIG. 15-16.—Timing diagram of this engine.

15-4. Cylinder Performance. *Compression.*—The compression line does not differ from that of four-stroke engines. Since the exhaust ports or valves are closed about 20 per cent after the dead center, the actual compression ratio is lowered by almost the same amount; however, the cylinder pressure at the beginning of the compression being higher than the outside pressure, as against a vacuum in four-stroke engines, this counteracts considerably the effect of the lower compression ratio. As a result, the nominal compression ratio, *i.e.*, r referred to the piston position at the dead centers, must be only slightly, 5 to 10 per cent, higher than in four-stroke engines.

Combustion and *expansion* lines do not differ at all from those of four-stroke engines.

Exhaust.—The exhaust and scavenge take place when the piston is near the dead center, and only a small portion of the work of expansion is lost. In spite of the fact that the exhaust release occurs about 60° of the crank travel before the dead center and the exhaust period occupies about one-third of each revolution.

The cylinder pressure at the moment of release at full load is about 35 to 50 psig. The period required for the pressure to fall to the scavenge pressure of about 2 to 4 psig depends upon the port area and engine speed and may vary from 15 to about 25° With these data it can be easily computed that about 50 per cent of the gases, by weight, have escaped, and the drop of the absolute temperature of the remaining gases, assuming an adiabatic expansion, being less than 30 per cent, the density of the remaining gases is much lower than that of the scavenge air.

Example 15-1.—Find the amount and density of the gases remaining in the cylinder of a 21½-in. × 31-in. two-stroke compression-ignition oil engine at the moment when scavenging starts. The engine has a nominal compression ratio of 13.5: 1; the exhaust release occurs 59° before dead center, and the scavenging starts 19° later; the cylinder pressure at the point of release is 43 psig, and the scavenge pressure is 3.0 psig; the engine develops 430 bhp/cyl at 225 rpm and uses 0.395 lb/hp-hr of fuel oil; the connecting-rod-stroke ratio is 2.5.

The weight of the gases at the moment of pressure release is equal to the weight of the charge at the beginning of the compression plus the weight of the injected fuel.

First must be found the volume of the charge: analytically or graphically the piston travel at 59° crank position from the above given data is found as 20.5 per cent; the length of the compression space assumed as being cylindrical is 31/(13.5 − 1) = 2.48 in.; therefore

$$V = 0.7854 \times 21.5^2[31 \times (1 - 0.205) + 2.48]/1728 = 5.705 \text{ cu ft}$$

The temperature of the charge, scavenge air mixed with residual gases, will be about 200 F; the pressure will be about 1 psi below the scavenge pressure, or 14.7 + (3 − 1) = 16.7 psia. Thus the weight of the charge will be, assuming R the same as for pure air,

$$W_a = \frac{16.7 \times 144 \times 5.705}{53.35 \times 660} = 0.3895 \text{ lb/cycle}$$

Amount of fuel injected per cycle

$$W_f = 430 \times 0.395/(60 \times 225) = 0.0126 \text{ lb/cycle}$$

The weight of the charge

$$W_{ch} = 0.3895 + 0.0126 = 0.4111 \text{ lb/cycle}$$

From the characteristic equation, at the point of release,

$$T_r = \frac{57.7 \times 144 \times 5.705}{0.4111 \times 53.35} = 2160 \text{ R}$$

At 40° crank position the piston is 9 per cent from the dead center and the corresponding volume is

$$V_2 = 0.7854 \times 21.5^2[31(1 - 0.09) + 2.48]/1728 = 6.45 \text{ cu ft}$$

The temperature, assuming an adiabatic expansion to 17.7 psia, is

$$T_2 = 2160(17.7/57.7)^{0.286} = 1542 \text{ R}$$

and

$$W_2 = 17.7 \times 144 \times 6.45/(53.35 \times 1542) = 0.200 \text{ lb}$$

Therefore the amount which has escaped is 0.4111 − 0.200 = 0.2111 lb, or slightly more than 50 per cent.

The density of the scavenge air, if its $T = 660$ R, is

$$w = 17.7 \times 144/(53.35 \times 660) = 0.0723 \text{ lb/cu ft}$$

and the density of the gases to be scavenged is

$$w = 0.200/6.45 = 0.031 \text{ lb/cu ft}$$

or about 43 per cent of the scavenge air.

Scavenging.—The instant at which admission of scavenge air starts is timed to coincide with that at which the cylinder pressure becomes equal to or slightly below the scavenge-air pressure. The incoming air is supposed to sweep the remaining burned gases before it and to fill the cylinder with a charge of pure air. However, a certain amount of mixing between the incoming air and the outgoing burned gases always exists. In addition, with the return flow, some short-circuiting of scavenge air to the exhaust ports takes place.

In order to counteract these processes and particularly in order to allow the first portion of the scavenge air, which is always fouled to a certain extent by burned gases, to be removed into the exhaust, an excess of scavenge air is provided. The average ratio of scavenge-pump displacement to cylinder displacement is usually about 1.4, sometimes up to 1.6. However, even with such an excess of air the fresh charge will contain from 5 to 20 per cent burned gases, depending upon the method of scavenging, the size and shape of the admission ports, and the scavenge pressure.

15-5. Scavenge Efficiency.—A good picture of the events in the cylinder of a two-stroke engine may be obtained from Fig. 15-17: W_p, in pounds, is the weight of air delivered by the scavenge pump; W_a is the weight of air and W_r that of residual gases mixed with air and left in the cylinder when the scavenge ports are closed; W_f is the weight of fuel admitted; W_b is the weight of gases that get into the exhaust, blown by the piston; W_d is the weight of the air corresponding to the piston displacement, either at standard conditions or at the pressure of the scavenge air, depending upon the basis of comparison; W_c is the weight of the air, at the same conditions, corresponding to the compression space. With these designations the scavenge efficiency η_{sc} is understood to be[1,2,3]

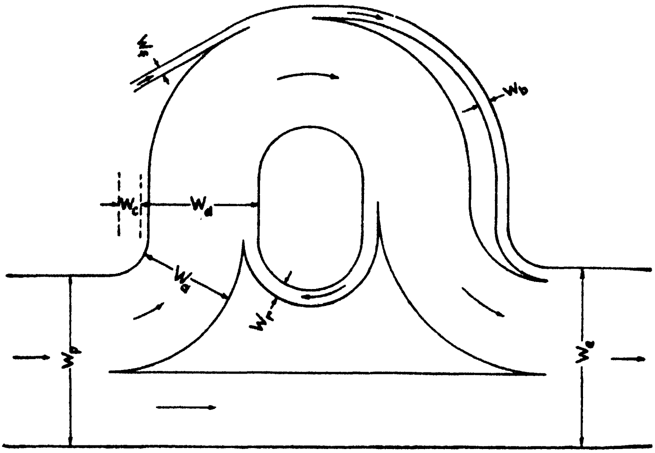

FIG. 15-17.—Sankey diagram of the air and gas flow in a two-stroke engine.

$$\eta_{sc} = W_a/(W_d + W_c) \qquad (15\text{-}1)$$

Scavenge efficiency has the same meaning for two-stroke engines as charge efficiency has for four-stroke engines. According to expression (15-1), the mean indicated pressure of a two-stroke engine with a 100 per cent scavenge efficiency would be higher than that of a four-stroke engine with natural aspiration having a 100 per cent charge efficiency, by the ratio $r/(r - 1)$, where r is the compression ratio of both engines.

In order to analyze the scavenging process of a two-stroke engine quantitatively, it is necessary to know, in addition to the scavenge efficiency, the amount of air which is delivered by the

[1] CURTIS, C., Improvements in Scavenging and Supercharging Two-cycle Diesel Engines, *Diesel Power*, September, 1933.

[2] The Scavenging of Two-cycle Engines, *Sulzer Tech. Review*, No. 4, 1933.

[3] ROGOWSKY, A. R., and BOUCHARD, C. L., Scavenging of a Piston-ported Two-stroke Cylinder, *NACA Tech. Notes*, 674, 1938.

scavenge pump at this efficiency. The ratio of the amount delivered by the pump to the cylinder content can be called scavenge factor K_{sc}. With the designations of Fig. 15-17

$$K_{sc} = W_p/(W_d + W_c) \tag{15-2}$$

Instead of K_{sc}, the value $K_{sc} - 1$ may be used and called the *excess air coefficient*

$$e = W_p/(W_d + W_c) - 1 \tag{15-3}$$

The excess air coefficient as found in commercial engines is about 0.3 to 0.5, meaning a scavenge factor K_{sc} of 1.3 to 1.5.

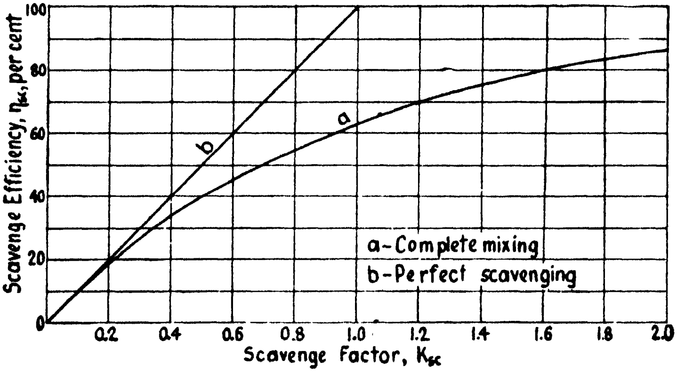

FIG. 15-18.—Effect of scavenge factor on scavenge efficiency.

Interesting information about the scavenging process may be obtained from a study of the hypothetical case of a cylinder containing a gas B into which a gas A is introduced so slowly that mixing of the two gases is complete at all times.[1] During this process the mixture of the gases is discharged to the exhaust at the same volume rate as gas A is admitted. If x is the proportion of gas A in the cylinder, then

$$dx = (1 - x)\,dv/V \tag{15-4}$$

where v is the volume of gas A admitted and $V = V_d + V_c$ is the total volume of the cylinder. Integration of equation (15-4) gives

$$x = 1 - 1/e^{\frac{v}{V}} \tag{15-5}$$

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[1] TAYLOR, C. F., and TAYLOR, E. S. "Internal Combustion Engines," p. 251, International Textbook Company, Scranton, Pa., 1938.

If the temperatures and pressures of the gases A and B are equal, x will be the scavenge efficiency and $v/V = W_p/(W_a + W_c)$, the scavenge factor. Curve a, Fig. 15-18, is a plot of scavenge efficiency against scavenge factor for this case.

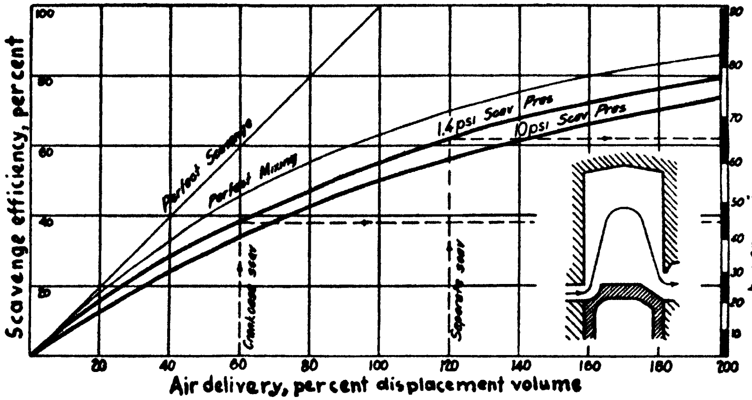

FIG. 15-19.—Scavenge efficiency of cross-scavenging.

Another hypothetical case is when a gas A is admitted into gas B and the latter is discharged to the exhaust at the same rate as A is admitted, without any mixing, line b. In an actual two-stroke engine the curve will run somewhere between a and b. However, if some air is short-circuited from the scavenge port directly to the exhaust port, as may happen with cross-scavenging, Fig. 15-1a, then the actual curve may run even below curve a.

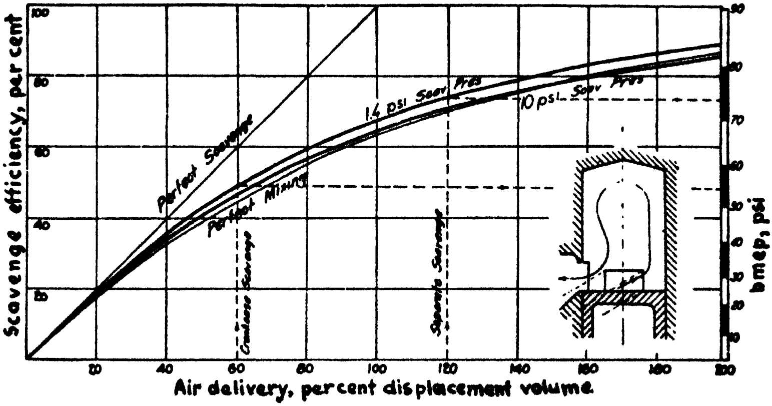

FIG. 15-20.—Scavenge efficiency of loop scavenging.

Using the same system of coordinates, scavenge efficiencies found experimentally were plotted together with the curves *a* and *b* for the three main types of scavenging, Figs. 15-19, 15-20, and 15-21.[1] The efficiency curves show: first, the relatively poor scavenging obtained with cross-scavenging and excellent scavenging obtained with uniflow scavenging and second, how an increase of the scavenge pressure, for example, from 1.4 to 10 psi, decreases the scavenge efficiency, evidently because of a greater formation of turbulences.

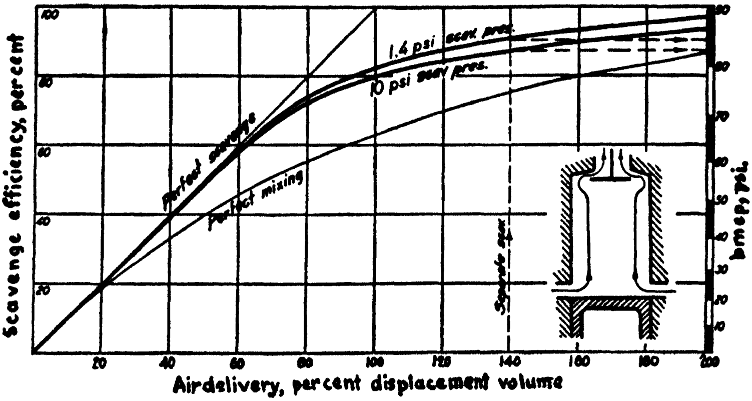

FIG. 15-21.—Scavenge efficiency of uniflow scavenging.

Mean effective pressure depends to a great degree on η_{se} and can be estimated by means of Figs. 15-19 to 15-21. The scale on the right of these diagrams serves to find the probable mean effective pressure which can be obtained with the corresponding type of scavenging. The relation between the amount of air available at different scavenge efficiencies and the obtained mean effective pressure was calculated with the following assumptions:

1. The cylinder charge at standard conditions is equal to the displacement volume multiplied by the corresponding scavenge efficiency, $W_d \times w \times \eta_{se}$.
2. The theoretical air-fuel ratio was taken as 14.6:1, which corresponds to a typical fuel oil, such as $C_{16}H_{30}$, Table 4-2.
3. The excess air *e* was assumed as 60 per cent, meaning an actual air-fuel ratio 14.6 × 1.6 = 23.3:1.

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[1] SCHWEITZER, P. H., Porting of Two-stroke-cycle Diesel Engines, *Diesel Power*, vol. 20, No. 5, p. 477, May, 1942.

4. The fuel consumption was assumed as 0.5 lb/hp-hr, which is a conservative value for two-stroke compression-ignition oil engines.

If an engine is not supercharged, the scavenge ports are closed before the exhaust ports; the values of mep read from the diagrams should be multiplied by about 0.70 to 0.75, depending upon the timing and port arrangement.

If the fuel consumption is lower than 0.5 lb/hp-hr, the mep read from the diagram must be increased in inverse proportion. Thus, for a fuel consumption of 0.405 lb/hp-hr the coefficient would be $0.5/0.405 = 1.235$. If the excess air e is other than 0.6, the mep can be increased 2.5 psi for each 10 per cent reduction of e or decreased 2.5 psi for each 10 per cent increase of e.

Example 15-2.—Compare the mean effective pressures that can be obtained with cross- and loop scavenging using: (*a*) crankcase compression and (*b*) a separate scavenge pump.

The volumetric efficiency of a crankcase pump may be assumed as 60 per cent; the scavenge pressure will be about 1.4 psi. For cross-scavenging, Fig. 15-19 gives, with these data, $\eta_{sc} = 0.38$ and a nominal $p_e = 45$ psi; since no supercharging exists with crankcase scavenging, the actual mep will be about

$$p_e = 45 \times 0.72 = 32.4 \text{ psi}$$

which is very close to what usually is found.

With a separate pump, a scavenge factor $K_{sc} = 1.2$ is a conservative figure. In this case, Fig. 15-19 gives $\eta_{sc} = 0.66$ and a nominal $p_e = 65$ psi, which actually can be obtained if a check valve is used, Fig. 15-4 or 15-6. If no check valve is used, the timing diagram corresponds to Fig. 15-3 and no supercharging is possible. In this case

$$p_e = 65 \times 0.72 = 46.8 \text{ psi}$$

With loop scavenging, Fig. 15-20 gives for crankcase scavenging $\eta_{sc} = 0.49$ and $p_e = 54$, or an actual mep

$$p_e = 54 \times 0.72 = 38.9 \text{ psi}$$

With a separate pump $p_e = 74$ psi, which corresponds to

$$p_e = 74 \times 0.72 = 53.3 \text{ psi}$$

Useful scavenge effect is the ratio of the amount of pure air that remains in the cylinder of a two-stroke engine at the beginning of compression to the total output of the scavenge pump. With the designations of Fig. 15-17

$$\text{u.sc.e.} = W_a/W_p \tag{15-6}$$

or with expressions (15-1) and (15-2)

$$\text{u.sc.e.} = \eta_{sc}/K_{sc} \tag{15-7}$$

From tests with commercial engines, useful scavenge effect was found to vary from about 0.75 down to 0.55, decreasing rapidly with an increase of the scavenge factor K_{sc}.

15-6. Scavenge Pumps.—The following schemes are used to obtain scavenge air:

(*a*) crankcase compression,
(*b*) pumps using the engine piston,
(*c*) separate reciprocating pumps,
(*d*) rotary pumps.

Crankcase compression is illustrated in Figs. 1-8 and 1-9. It is used in many smaller, up to about 70 hp/cylinder, American spark-ignition gas engines and compression-ignition oil engines. The volumetric efficiency of such a pump is low, from about 55 to 65 per cent.

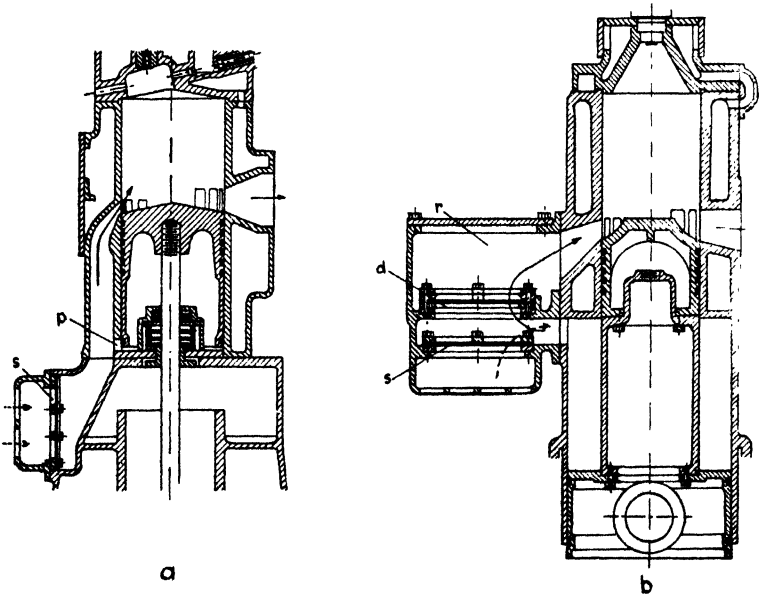

FIG. 15-22.—Reciprocating scavenge pumps.

A pump using the back of the engine piston is shown in Fig. 15-22*a*; *s* is the suction valve and *p*, ports to the pump cylinder. This pump type is used in Cooper-Bessemer and Superior gas and oil engines. While the volumetric efficiency of such a pump may be sufficiently high, it cannot give any excess air necessary for efficient scavenging.

A pump with a step piston, also called *differential piston,* Fig. 15-22*b*, permits the desired excess air but requires a sufficiently large air receiver *r*, as the end of the discharge stroke does not coincide with the scavenge period; *s* is the suction and *d* the discharge valve. Such a pump is used in Standard Fuel Oil and Weber gas and compression-ignition oil engines. Its drawback is in the heavy reciprocating masses, which make it unsuitable for higher speeds and powers.

A separate scavenge pump *A* built into a small engine is shown in Fig. 15-23. The piston *B* is driven from a pin fastened to the main connecting rod. This arrangement makes it possible to have any amount of excess scavenge air and also to have the proper timing of the air delivery. The engine is only slightly bulkier and heavier than a crankcase-scavenge engine having the same bore and stroke, but develops more power and thus actually is less bulky and lighter per rated horsepower. The dead corner space between the working cylinder and pump cylinder is adroitly utilized for placing the large exhaust-manifold muffler *H*, very desirable in a two-stroke engine.

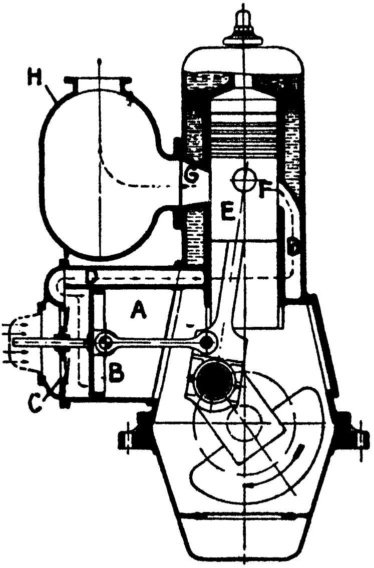

Fig. 15-23.—Small two-stroke oil engine with scavenge pump.

A double-acting scavenge pump as used in large engines is shown in Fig. 15-24. The pump is driven by a separate crank fastened to the end of the crankshaft; one pump cylinder usually serves all the cylinders of a multicylinder engine. In order to reduce the pump bore, the stroke is often made longer than the engine stroke.

High-speed engines preferably use rotary pumps, of the improved Roots type with three lobes, Fig. 1-17. The sides of

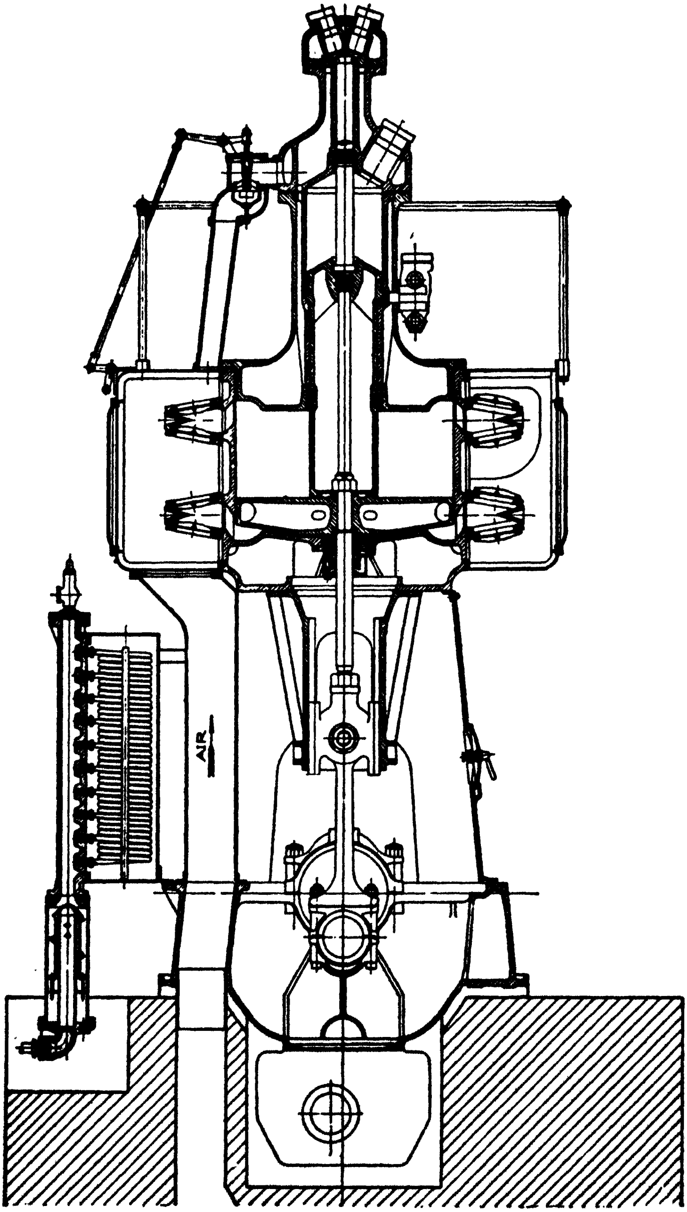

FIG. 15-24.—Double-acting reciprocating scavenge pump.
(*Courtesy of Nordberg Manufacturing Company.*)

the lobes are helical surfaces in order to give a uniform air motion and to reduce the air noise. Roots blowers are used also in some medium-speed large engines, such as the Busch-Sulzer.

15-7. Comparison of Two- and Four-stroke Engines.—From the previous discussion it is obvious that thermodynamically the two types of engines do not differ; the difference is purely mechanical.

The chief advantages of two-stroke engines, at least theoretically, are: one-half smaller piston displacement for a given power, meaning an engine practically one-half as heavy and less expensive; a flywheel about one-half as heavy for the same uniformity of rotation because of twice the number of the working strokes, giving a higher mechanical efficiency; and ease of supercharging.

However, the actual performance of two-stroke engines is not always as satisfactory as would be expected. The main drawback is in the difficulty of obtaining good scavenging, unless a large excess of scavenge air is used; poor scavenging means a reduction of the oxygen content in the charge, and consequently of the amount of fuel that can be burned; as a result considerably lower mean indicated pressures are obtained than theoretically expected. With crankcase scavenging they are even lower than in four-stroke engines.

TABLE 15-1.—BRAKE MEAN EFFECTIVE PRESSURES OF COMPRESSION-IGNITION TWO-STROKE OIL ENGINES

Type of scavenging	Range, psi	Average, psi
Crankcase scavenge engines	28– 40	35
Crosshead engines, straight piston	34– 48	40
Step-piston engines	40– 56	45
Return-flow scavenging, separate pump	53– 70	62
Uniflow engines with one piston	60–100	85
Uniflow engines, opposed pistons	70–120	90

Table 15-1 gives mean effective pressures as obtained in 121 modern commercial two-stroke compression-ignition oil engines.[1]

The expected increase of mechanical efficiency is not great, if any, and the fuel consumption per horsepower-hour is even slightly higher than in four-stroke engines, except in larger units.

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[1] SCHWEITZER, *op. cit.*, p. 401.

There are other difficulties, particularly with respect to building high-speed, small-bore two-stroke engines. Only the open combustion chamber seems to be practicable for two-stroke engines which in itself is not so desirable for smoothness and quietness of operation as indicated in Chap. 11. The problem of cooling the piston is complicated by the fact that the piston does not have the benefit of the second revolution when no heat is being generated in a four-stroke engine. The danger of overheat ing the piston and some other parts limits the maximum power of a two-stroke engine in order to maintain reasonable dependability and durability or requires oil cooling of the pistons. The balancing of the crankshaft, particularly for a high-speed four-cylinder engine, presents certain difficulties because the cranks are not in one plane. The same is true, although to a lesser degree, for six- and eight-cylinder engines.

The unbalance of the crankshaft makes it necessary to take the camshaft drive off the flywheel end of the crankshaft, which makes the drive less accessible.

Only engines with separate scavenge pumps and operating on one of the more perfect schemes give really good results. These engines with few exceptions are built in large units where complications of design are not so important and where lighter weight, more uniform rotation, and possibility of supercharging are particularly valuable.

In general, two-stroke engines are built as compression-ignition engines. Small- and medium-size engines, approximately up to 1000 hp, are built usually as four-stroke engines; above that size, two-stroke oil and gas engines begin to compete successfully with four-stroke engines until in sizes from 5000 hp up, built in double-acting or opposed-piston multicylinder units, they seem to push the four-stroke oil engines to the background. However, engines using uniflow scavenging, Fig. 15-15, are built in large numbers in sizes from 25 to 125 hp per cylinder, and in units up to 16 V-type cylinders.

15-8. Problems.—1. A two-stroke oil engine with port scavenging, Fig. 15-1, has an 8-in. bore, 11-in. stroke, a 25-in. long connecting rod, and runs at 360 rpm. Determine the length of the exhaust and scavenge ports if the exhaust release occurs 57° before dead center; scavenging begins 40° before dead center. Also determine the points of exhaust release and beginning of scavenging in percentage of engine stroke, and duration of exhaust and scavenging in seconds.

2. A two-stroke gas engine with valve scavenging according to Fig. 15-4 has a 12-in. bore, 16-in. stroke, and runs at 285 rpm. The exhaust-port length is 2¾ in.; the length of the connecting rod is 40 in.; the automatic scavenge valve opens 18° of crank travel after the exhaust release and closes 12° after the exhaust port is closed. Find: (*a*) the duration of exhaust and scavenging in seconds, and (*b*) the length of the scavenge ports.

3. A four-cylinder two-stroke engine with a combination valve-and-port scavenging, Fig. 15-6, has a 20-in. bore, 26-in. stroke, and runs at 220 rpm. The connecting rod is 60 in. long. The exhaust release occurs 60° of crank travel before the dead center, the scavenge port *b* begins to open 40° before the dead center, and port *a* is closed 25 per cent after the dead center. Determine the length of all three ports, *a*, *b*, and *e*, if the wall between *a* and *b* is ⅞ in. thick. Also find the duration in seconds of the exhaust and scavenging.

4. A M.A.N. double-acting two-stroke engine, scheme Fig. 15-8, has a 9-in. bore, 13⅜-in. stroke, and runs 750 rpm. The length of the connecting rod is 30 in.; the distance *a* from the edge of the scavenge port to the end of the piston is 1.9 in.; distance *b* is 4 in. Find the positions of the crank corresponding to the exhaust-port opening, scavenge-port opening, and scavenge and exhaust closing, also in per cent of the piston stroke for both cylinder ends and the duration of all events in seconds.

5. A Hamilton-M.A.N. double-acting engine, scheme Fig. 15-8, has a 27½-in. bore, a 47½-in. stroke, and runs at 100 rpm; the length of the connecting rod is 95 in. Find the length (*a*) of the exhaust and (*b*) of the scavenge ports, if the exhaust release occurs at 60° of the crank travel and the scavenging starts 25° later; the lands between the exhaust and scavenge ports are ⅝ in.; find the duration in seconds (*c*) of the exhaust and (*d*) of the scavenging, and in per cent of piston stroke the moment (*e*) of the exhaust release and (*f*) of the beginning of scavenging.

6. A Junkers aircraft oil engine operating on scheme Fig. 15-11 has a 3.15-in. bore by 7.9-in. + 7.9-in. stroke, and runs at 1000 rpm. The connecting rods are 21.7 in. long; the crank of the upper piston runs 12° ahead of the crank of the lower piston; the duration of the exhaust is 100° of the crank travel, the same for the scavenging. Determine: (*a*) the length of the exhaust and scavenge ports and (*b*) the duration in seconds of scavenging after closing of the exhaust.

7. A Junkers aircraft oil engine, Fig. 15-11, has a 4¾-in. bore, a 8¼-in. + 8¼-in. stroke, and runs at 1700 rpm. The connecting rods are 22½ in. long; the crank of the upper piston runs 13° ahead of the crank of the lower piston; the duration of the exhaust is 118° of the crank travel, the same for the scavenging. Determine: (*a*) the length of the exhaust and scavenge parts, (*b*) the loss of the piston stroke in per cent before the compression starts, and (*c*) the duration in seconds of scavenging after closing of the exhaust ports.

8. Find (*a*) the amount and (*b*) the density of the gases remaining in the cylinder of the engine of problem 3; the cylinder pressure at the point of release is 60 psia; the scavenge pressure is 3.3 psig; the mean effective pressure is 65 psi; and the engine uses 0.39 lb/hp-hr of fuel oil; the nominal compression ratio is 13.7:1; assume that the temperature of the charge at the beginning of the compression is 205 F.

9. For the engine of problems 3 and 8, find: (*a*) the scavenge efficiency, knowing that the double-acting scavenge pump, Fig. 15-24, has a bore of 32 in. and the same stroke as the working pistons; assume a dilution of 15 per cent; (*b*) find the scavenge efficiency by using Fig. 15-19.

10. For the engine discussed in preblems 3, 8, and 9, compute the probable average temperature of the exhaust gases in the exhaust pipe.

11. Determine the expected scavenge efficiencies and mean effective pressures for engines (*a*) with cross-scavenging and (*b*) using a separate pump with a 1.25 scavenge factor and a scavenge pressure of 3 psi. Use the timing diagram: (*a*) of Fig. 15-3 and (*b*) of Fig. 15-5.

12. Work problem 11 for an engine using loop scavenging.

13. Determine the expected scavenge efficiencies and mean effective pressures for an engine with a single piston and uniflow scavenging for a scavenge pressure of 3.75 psi if the scavenge factor is: (*a*) 1.3, (*b*) 1.4, and (*c*) 1.5; (*d*) plot calculated mep against scavenge factor.

14. Work problem 13 for a scavenge pressure of 5 psi and scavenge factors of 1.25, 1.40, and 1.55.

CHAPTER 16

ENGINE EFFICIENCIES

16-1. Enumeration.—The following efficiencies are encountered when dealing with the theory, design, and operation of various internal-combustion engines:

1. Thermodynamic, or air-standard efficiency.
2. Ideal efficiency.
3. Indicated thermal efficiency.
4. Relative efficiency.
5. Mechanical efficiency.
6. Brake, or over-all, thermal efficiency.
7. Combustion efficiency.
8. Volumetric efficiency.
9. Charge efficiency.
10. Scavenge efficiency.

Several of these efficiencies have already been discussed and are enumerated here chiefly to give a more complete picture.

Numerically, efficiencies are expressed either in decimal fractions or in per cent.

16-2. Theoretical Efficiencies.—*Thermodynamic efficiency* is a function of the compression ratio and method of combustion. Since theoretical cycles are based on air as working substance, their efficiency is also called *air-standard efficiency.*

Ideal efficiency is the efficiency of an ideal engine, which is without heat losses to the walls but uses the same working substances as do actual engines. In computing the ideal efficiency η_{id}, variable specific heats and conditions of chemical equilibrium at elevated temperatures are taken into account. The determination of ideal efficiencies is considerably simplified by combustion charts, Figs. 4-2 to 4-9. A still simpler method is to use the curves as given in Figs. 5-10 and 5-11.

16-3. Indicated Thermal Efficiency.—This efficiency is designated by η_t and is the ratio of the heat equivalent of indicated work performed by the engine to the heat consumed during the same time,

$$\eta_i = 2545N_i/WQ \tag{16-1}$$

where 2545 is the equivalent of 1 hp-hr in Btu, N_i is the indicated horsepower developed, W the fuel consumption in cubic feet per hour for gas or pounds per hour for liquid or solid fuel, and Q the heat value of the fuel in Btu per cubic foot or per pound, as the case may be.

The indicated horsepower N_i of a two-stroke engine is represented by the total area of its indicator card. The N_i of a four-stroke engine is represented by the area 1-2-3-4-1, Fig. 16-1, which corresponds to the theoretical diagram. Often the N_i of a four-stroke engine is computed by using a planimeter and by following the contour from point 1 over 2, 3, 4, and 5 back to 1. In this case, loop 6-5-1-6 becomes negative and is subtracted automatically from the positive loop 6-2-3-4-6. Strictly speaking this procedure is not correct; the area 4-5-1-4 is the work of charging the cylinder and should not be deducted from the indicated work, just as the scavenge-pump work is not deducted from the indicated work of a two-stroke engine.

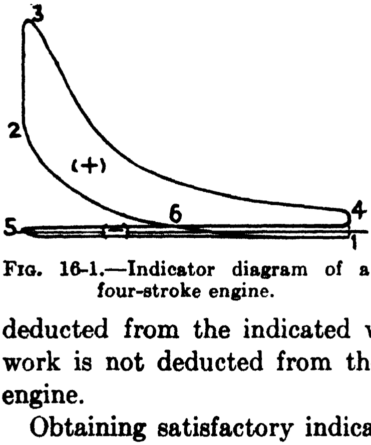

FIG. 16-1.—Indicator diagram of a four-stroke engine.

Obtaining satisfactory indicator diagrams from internal-combustion engines with the usual piston-type indicator is rather difficult for several reasons: First, the opening of the indicator cock increases the compression space appreciably and thus influences both compression and combustion, particularly in small high-compression engines; second, the indicator piston cannot follow accurately the rapid pressure changes taking place at high speeds; third, the inertia of the moving indicator masses tends to distort the indicator diagram; finally, the absence of a cross-head in most engines and the enclosed-crankcase construction make it more difficult to build a correct reduction mechanism. Indicators of other types are available, but they are rather expensive and require a refined technique to obtain accurate diagrams, The simplest method of finding the approximate indicated work of high-speed engines is to add to the useful, or brake, horsepower N_b of the engine the mechanical losses N_r, found by driving, "motoring," the engine with an electric motor at conditions approaching as nearly as possible the operating conditions,

$$N_i = N_b + N_r \tag{16-2}$$

Analyzing the operation of an engine, it will be found that N_r depends to a great extent on the speed of the engine and very little upon its useful work N_b.

In modern engines an indicated efficiency η_i of almost 30 per cent is obtained with gas- and gasoline-spark-ignition engines having a moderate compression and η_i as high as 40 per cent and even more with high-compression oil engines.

16-4. Relative efficiency η_r is the ratio of the indicated efficiency to the ideal efficiency

$$\eta_r = \eta_i/\eta_{id} \tag{16-3}$$

and shows how close the actual engine comes to the theoretically possible performance. The ideal efficiency must be taken from Fig. 5-10 to 5-12 or calculated by means of combustion charts.

Relative efficiency obviously is influenced by the same factors as indicated efficiency η_i. In present engines η_r attains values from 0.85 to 0.95 with theoretical air or excess air[1] and decreases rapidly with insufficient air to about 0.77 with 90 per cent air.[2]

16-5. Mechanical efficiency is the ratio of available, or brake, horsepower N_b to the indicated horsepower N_i

$$\eta_m = N_b/N_i \tag{16-4}$$

where N_i often is found from expression (16-2).

The mechanical losses N_r can be divided into four main groups:

1. Friction losses, as in pistons, bearings, gears, and valve mechanisms including opening of the valves.
2. Work of charging—absorbed during the exhaust and suction strokes in four-stroke engines, or by the scavenge pumps in two-stroke engines.
3. Ventilating action of the flywheel.
4. Power absorbed by auxiliaries, such as lubricating pumps, water-circulating pumps, fuel pumps, air compressors in air-

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[1] *Z. Ver deut. Ing.*, vol. 85, No. 5, p. 116, 1941.

[2] LICHTY, L. C., "Internal-Combustion Engines," 5th ed., pp. 115, 116, McGraw-Hill Book Company, Inc., New York, 1939.

injection oil engines, magneto and distributor drives, radiator fans, and electric generators for battery charging.

In general, the friction losses, group 1, vary from 7 to 10 per cent of the indicated power; the work of charging varies from 2 to 6 per cent; the ventilating losses vary from 1 to 3 per cent; the power to drive auxiliaries varies from 1 to 9 per cent. The sum of all mechanical losses varies from 11 to about 30 per cent, and the mechanical efficiencies vary accordingly between 72 and 89 per cent.

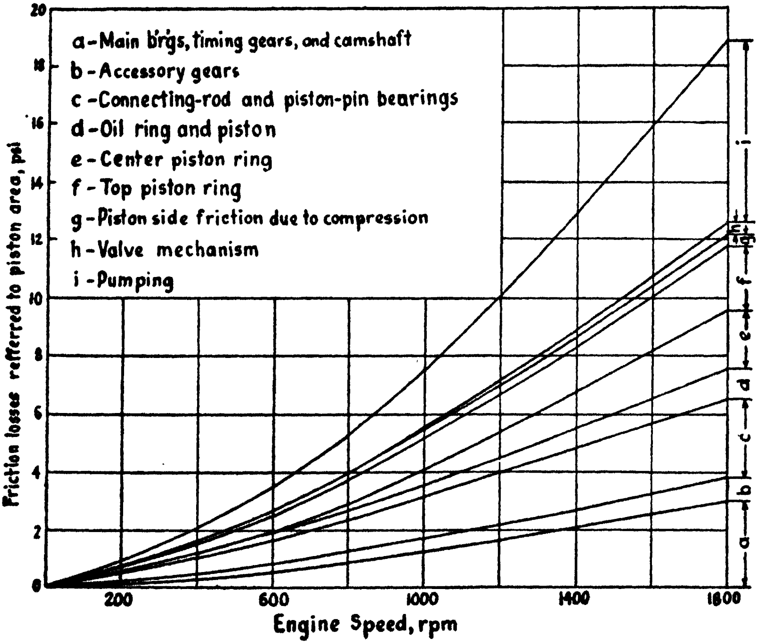

FIG. 16-2.—Friction losses in a four-stroke truck engine.

A convenient method for investigating the losses in groups 1 and 2 is to express them as negative pressure p_r, psi, on the engine piston area and from this pressure to compute the corresponding loss N_r, hp.

The pumping and friction losses depend on the following engine characteristics: compression ratio, cycle of operation, single- or double-acting, size and number of cylinders, and piston speed, as well as on the precision used in manufacturing and assembling the parts. Figure 16-2 gives a good picture of the relative values of pumping and friction losses and a breakdown of the latter into the main components.[1]

These losses depend also on the jacket-cooling water, method, quality, and quantity of lubrication, and on engine speed. Figure 16-3 shows the influence of the water and oil temperatures upon the various items.[2] In order to avoid misinterpretation it should be noted that the actual losses through friction of the piston, piston rings, and of the valve mechanism decrease with an increase of the cooling-water temperature due to the lowering of the lubricating oil viscosity. The pumping losses, on the other hand, increase with the water temperature as the volumes of both the exhaust and the intake gases increase. The bearing losses are practically independent of the water temperature. The lubricating oil temperature affects chiefly the bearing losses.

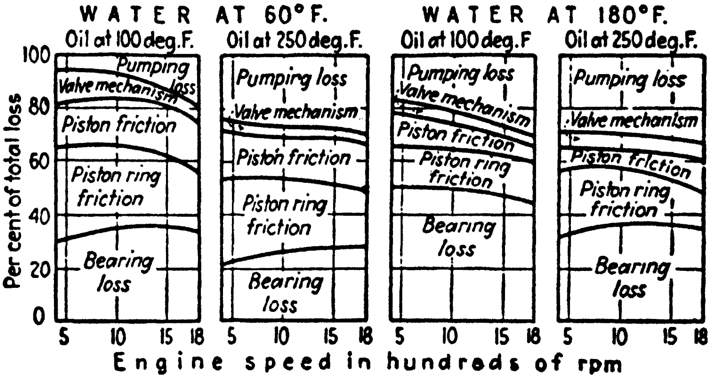

FIG. 16-3.—Effect of oil and water temperatures on distribution of friction losses.

Speed Influence.—The influence of speed upon the various losses can be seen from Fig. 16-2, whereas Fig. 16-4 presents the total friction, groups 1 and 2, for several typical engines.[3] In general, well-designed aircraft engines have relatively low friction losses. Automobile engines have higher charging losses and considerably higher friction losses because of longer and closer fitted pistons and lower cylinder temperatures. Stationary, low- and medium-speed engines occupy an intermediate position.

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[1] Based on data from Lichty, *op. cit.*, p. 452, and additional data given by Professor Lichty.

[2] LICHTY, *op. cit.*, p. 453.

[3] TAYLOR and TAYLOR, *op. cit.*, p. 204.

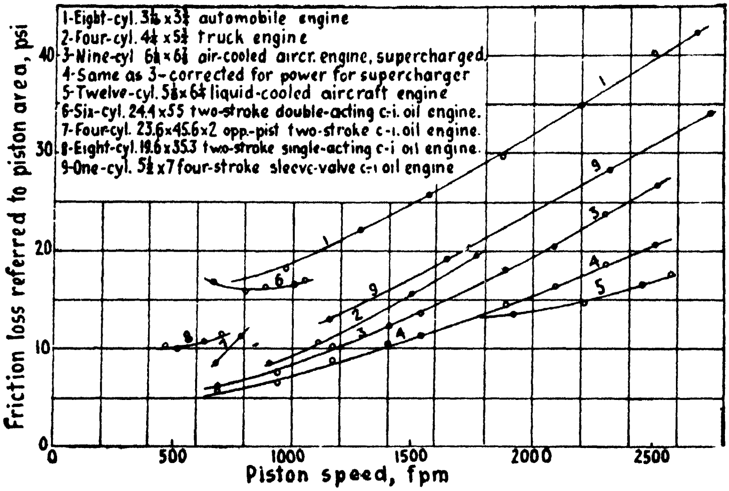

FIG. 16-4.—Effect of piston speed on friction loss for several typical engines.

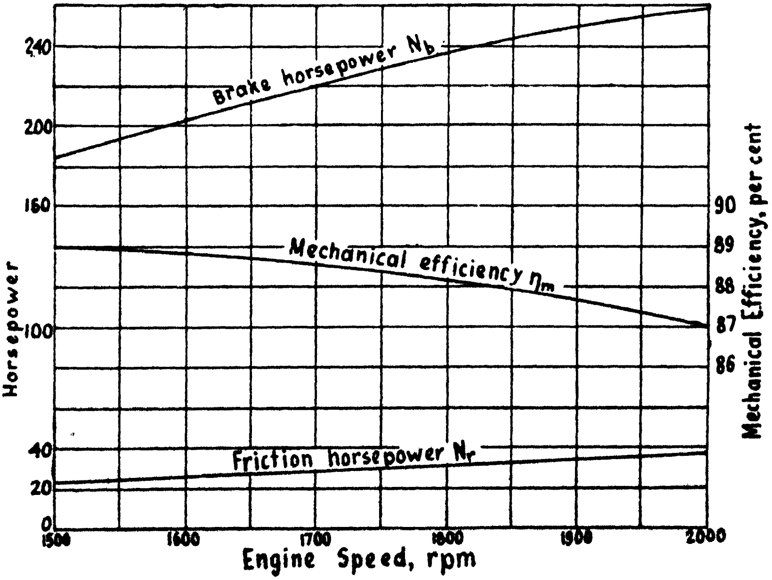

FIG. 16-5.—Effect of speed on mechanical efficiency of an aircraft engine.

For a given engine the mechanical efficiency decreases with speed, as can be seen from Fig. 16-5, for an aircraft engine, and from Fig. 16-6, for an automobile engine.

Altitude.—Influence of altitude, as already discussed, results in a decrease of the maximum output. The pumping losses and ventilating action of the flywheel also decrease with a decrease of the air density; however the mechanical losses, groups 1 and 4, remain practically unchanged. Therefore the mechanical efficiency of an engine decreases with the altitude. Taking into account that the air densities are proportional to pressures, the mechanical efficiency η_m' at an altitude h and pressure p can be found from the mechanical efficiency η_m at sea level as[1]

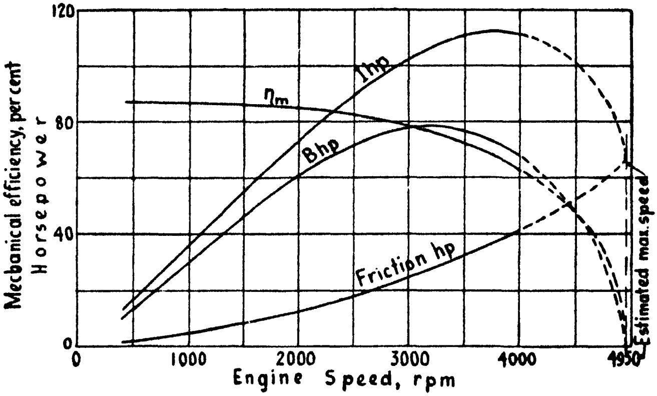

Fig. 16-6.—Performance curves of an automobile engine.

$$\eta_m' = \eta_m - 0.7(1 - \eta_m)[(14.7/p)^{0.81} - 1] \qquad (16\text{-}5)$$

This formula agrees well with NACA data.[2]

Load.—The mechanical efficiency of an engine is a function of the load, falling off with the decrease of load. As a first approximation, as in the case of steam engines, the losses N_r can be considered to be independent of the load; in other words the losses are equal to the indicated horsepower of the engine when running idle. This gives for the mechanical efficiency at any load N_b' the simple formula

$$\eta_m = N_b'/(N_b' + N_r) \qquad (16\text{-}6)$$

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[1] Devillers, R., "Le Moteur d'explosion," p. 158, Dunod, Paris, 1935.

[2] *NACA Tech. Notes* 619, 1937, p. 7.

However, reliable values for η_m can be obtained only from actual tests.

Finally, Fig. 16-7 gives typical curves of mechanical efficiency for representative types of engines at different loads. These data can be used for estimating the mechanical efficiencies of engines of similar characteristics.

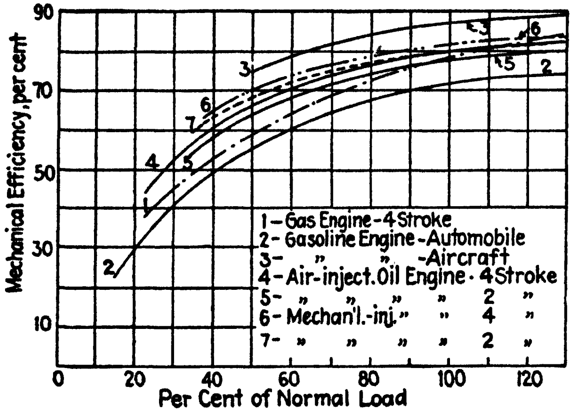

FIG. 16-7.—Typical curves of mechanical efficiency.

16-6. Brake thermal efficiency, also called *over-all efficiency,* is the ratio of the heat equivalent of the brake, or useful, horsepower N_b, developed by an engine and available on its crankshaft, to the heat Q consumed during the same time,

$$\eta_b = 2545N_b/WQ \qquad (16\text{-}7)$$

Substituting for N_b its value from expression (16-4) and for N_i its value from expression (16-1), gives

$$\eta_b = \eta_i\eta_m \qquad (16\text{-}8)$$

and with expression (16-3),

$$\eta_b = \eta_{id}\eta_r\eta_m \qquad (16\text{-}9)$$

The influence of the compression ratio r upon η_i, η_m, and their product η_b is shown in Fig. 16-8 for a theoretical engine with a variable compression ratio. The indicated thermal efficiency η_i rises with the increase of r; however, the curve of η_m declines and as a result the curve of η_b at first rises, up to about $r = 11$, then stays constant, up to $r = 12$, and after this begins to drop, showing that a further increase of r is useless so far as over-all efficiency η_b is concerned. The curve p_c shows the compression pressures with different r, and the curve p_r shows the corresponding friction pressures, $p_r = p_i - p_e$.

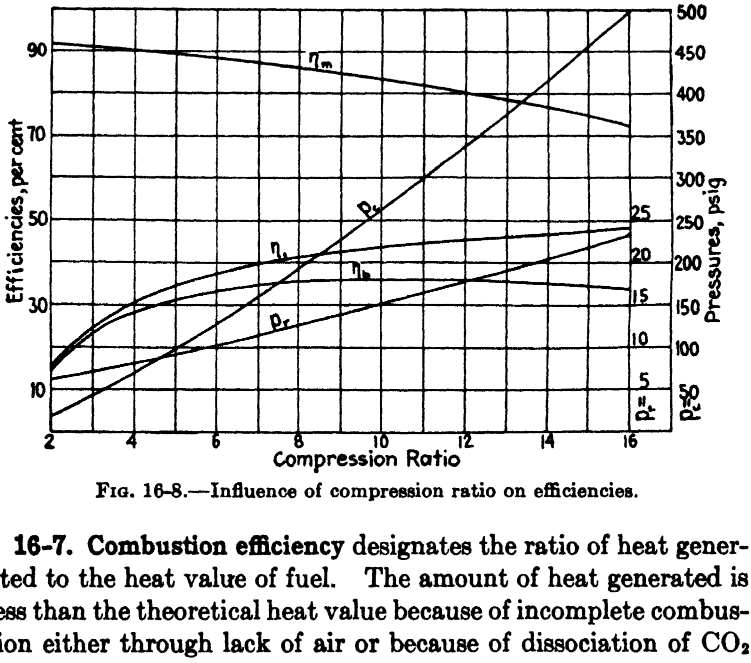

FIG. 16-8.—Influence of compression ratio on efficiencies.

16-7. Combustion efficiency designates the ratio of heat generated to the heat value of fuel. The amount of heat generated is less than the theoretical heat value because of incomplete combustion either through lack of air or because of dissociation of CO_2 and H_2O at high temperatures. Incomplete combustion is one of the reasons why the indicated efficiency η_i is lower than the ideal efficiency η_{id}. Sufficient excess air and proper turbulence can and actually do give combustion efficiencies equal to 1, whereas operating an engine with an insufficient amount of air or poor mixing may result in a combustion efficiency of 0.95 or even less.

16-8. Charge efficiency was discussed in Sec. 12-8, and the following expression was deduced in connection with Fig. 12-5:

$$\eta_{ch} = T_a[rp_d/T_d - (p_c/T_e)(n_1/n_2)]/[p_a(r - 1)] \qquad [12\text{-}20]$$

The charge efficiency shows how well the piston displacement of a four-stroke engine is utilized.

If the charge efficiency is referred not to the outside conditions p_a and t_a but to standard conditions of 14.7 psia and 60 F, expression (12-20), taking into consideration the meaning of the suction pressure p_d, becomes

$$\eta_{ch}' = \frac{520}{14.7(r-1)}\left\{\frac{r[14.7-(p_a-p_d)]}{520+t_m+t_h} - \frac{[14.7+(p_c-p_a)n_1]}{T_r n_2}\right\} \qquad (16\text{-}10)$$

where t_m is found by equation (12-6) and t_h by equation (12-13).

However, it should be remembered that expressions (12-20) and (16-10) have different meanings: the first shows the influence of actual conditions both inside and outside of the engine upon its performance, whereas the second refers to standard conditions and is used in computations when comparing different engines or tests of the same engine under different outside conditions.

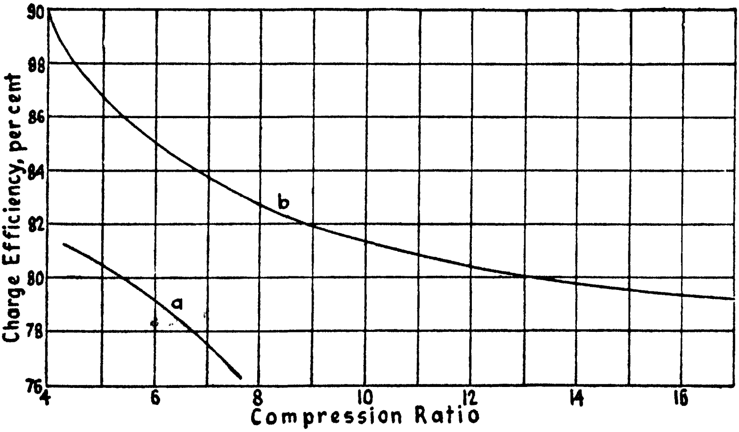

FIG. 16-9.—Effect of compression ratio on charge efficiency.

The main factors which influence the numerical value of both can be listed as follows:

1. The compression ratio r.
2. The resistance offered to the air-fuel charge during its passage through the induction system into the cylinder.
3. The amount of heat picked up by the air-fuel charge during its passage to the cylinder and from the cylinder itself.
4. The timing of the engine.

The influence of several of these factors can be seen readily from equation (16-10).

With an increase of r, all other conditions being equal, η_{ch} and η_{ch}' decrease. Curve a, Fig. 16-9, shows the influence as found experimentally for a gas engine. Curve *b* gives the influence computed theoretically.

The resistance in the induction system causes a certain back pressure, $p_c - p_a$, and a certain vacuum, $p_a - p_d$. An increase of either or both values lowers η_{ch} and η_{ch}'. In a given engine the resistance in the intake system increases with the engine speed and lowers η_{ch}. Figure 16-10 confirms this in general but shows also that a certain speed exists at which the flow resistance is a minimum and a further speed reduction is not beneficial.

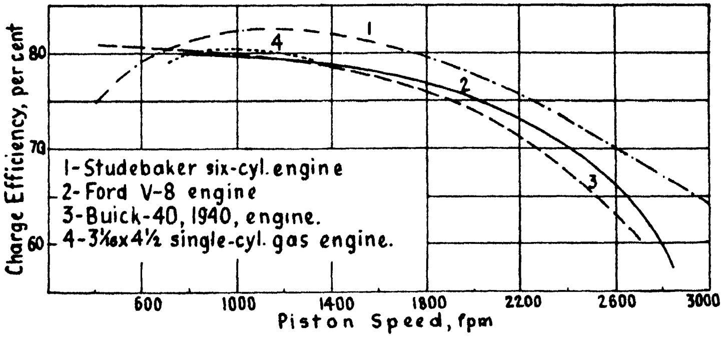

FIG. 16-10.—Effect of piston speed on charge efficiency of spark-ignition engines.

The amount of heat picked up by the charge raises its temperature by $t_m + t_h$ and decreases its density, thus lowering η_{ch}.

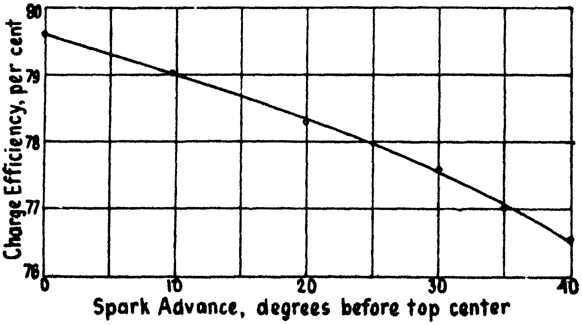

FIG. 16-11.—Effect of spark advance on charge efficiency.

The timing refers to the valves and to ignition. For each engine there exists a timing of the valves when they give a highest η_{ch} by reducing the resistance or by taking advantage of the inertia of flowing gases. The character of the ignition timing can be seen from Fig. 16-11.[1] With advanced ignition, the temperature of the gases and cylinder walls goes up, more heat is given up to the incoming charge; this results in a decrease of the charge efficiency.

A more accurate definition, useful if combustion occurs with an excess of air, is given by expression (12-21). While it is more difficult to present it as a function of temperatures, pressures, and compression ratio, in practice computation of η_{ch} by expression (12-21) is not difficult.

Volumetric efficiency, already discussed in Sec. 12-2, is the ratio of the volume of the fresh charge at outside pressure to the piston displacement,

$$\eta_v = V_{ch}/V_s \qquad [12\text{-}1]$$

The chief interest in the volumetric efficiency is in its influence upon the charge efficiency and for computing the charge temperature, expression (12-5).

16-9. Two-stroke Engines.—In dealing with two-stroke engines a concept similar to charge efficiency is useful but, to avoid confusion, is given a different name.

Scavenge efficiency η_{sc} was already defined as the ratio of the amount of air or gas-air mixture which remains in the cylinder of a two-stroke engine at the actual beginning of the compression to the product of the total cylinder volume and air density of the inlet.

The actual scavenge efficiency of an engine in operation can be determined by measuring the amounts of air and fuel furnished and by analyzing samples of the charge at the beginning of the compression and of the exhaust gases escaping during the lead angle.

From such tests it was found that η_{sc} varies from about 0.40, in engines with simple port scavenging and crankcase pump, up to 0.95, in engines with separate scavenge pumps with a sufficient excess of scavenge air.

In designing a two-stroke engine the scavenge efficiency may be estimated with a sufficient accuracy from the curves of Figs. 15-19 to 15-21.

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[1] VOGT, C. J., Characteristics of Internal Combustion Engines When Operating with Butane-air Mixtures, *Oil Gas J.*, Nov. 14, 1935.

16-10. Problems.—1. Find the indicated thermal efficiency of a gas engine which develops at full load 80 ihp, using 675 cu ft gas per hour; the heat value of the gas is 970 Btu/cu ft.

2. Find the relative efficiency for the engine in problem 1 if it has a compression ratio $r = 5.0$ and operates on 120 per cent theoretical air.

3. Find the indicated thermal efficiency at the rated load of a blast-furnace gas engine which at full load develops 2000 ihp, and uses 152,500 cu ft gas per hour; the heat value of the gas is 105 Btu/cu ft.

4. Find the relative efficiency for the engine in problem 3 if it has a compression ratio $r = 7.5$ and operates on 125 per cent theoretical air.

5. Find the guaranteed indicated thermal efficiency of an oil engine which must develop 4500 ihp at full load using 0.36 lb/ihp-hr fuel 20° API gravitv.

6. Find the relative efficiency for the engine in problem 5; $r = 14.0$; $e = 0.75$.

7. If the load of the engine of problem 1 is 64 bhp, plot a curve of the mechanical efficiency against the load factor assuming constant mechanical losses, expressions (16-2) and (16-6).

8. If the load of the engine of problem 3 is 1650 bhp and the mechanical losses are considered as remaining constant from ¼ to 1⅛ load, plot a curve of the mechanical efficiency against the load factor.

9. Assuming that the load of the engine in problem 5 is 3700 bhp and that the mechanical losses remain constant from 0.2 to 1.1 load, plot a curve of the mechanical efficiency against the load factor.

10. (*a*) Find the brake thermal efficiency for the engine in problems 1 and 7 for full-, three-quarter-, and half-load conditions, assuming the same indicated specific fuel consumption; (*b*) plot a curve of efficiency against brake load factor.

11. (*a*) Find the brake thermal efficiency for the engine in problems 3 and 8 for 1⅛, ¼, ¾, and ½-load conditions, assuming the same indicated specific fuel consumption; (*b*) plot a curve of efficiency against brake load factor.

12. (*a*) Calculate the brake thermal efficiency for the engine in problems 5 and 9 for 1.1, 1.0, 0.75, and 0.5-load conditions, assuming the same indicated specific fuel consumption; (*b*) plot a curve of efficiency against brake load factor.

13. Find the mechanical efficiency at full-load at altitudes of 5,000 and 12,000 ft for the engine discussed in problems 1 and 7.

14. (*a*) Find the mechanical efficiency at full load at altitudes of 6,000 and 11,500 ft for the engine discussed in problems 5 and 9; (*b*) calculate the approximate mechanical efficiency at three-quarter loads at the above altitudes.

15. (*a*) Find the mechanical efficiencies at full load at altitudes of 10,000, 15,000, and 25,000 ft above sea level for an aircraft engine which at sea level has a mechanical efficiency of 0.885; (*b*) plot a curve of efficiency against altitudes.

16. (*a*) Find the mechanical efficiencies at full load at altitudes of 12,500, 17,500, and 22,500 ft for an aircraft engine which at sea level has a mechanical efficiency of 0.895; (*b*) plot a curve of efficiency against altitude.

CHAPTER 17

ENGINE DESIGN

17-1. Selection of Type. *Cycle.*—Whether to build a two- or a four-stroke-cycle engine is often not so easy to decide, as each type has its advantages and disadvantages. Personal preference and previous experience have a great deal to do with the selection of type. However, there are certain cases when the selection is more or less predetermined by the general conditions. A small oil engine, for which the fuel economy is not so important as simplicity, should be built on the two-stroke cycle with crankcase scavenging. Large double-acting oil engines, particularly where reduction of weight is an advantage, as in marine engines, also are built as two-stroke-cycle engines. They are equipped with elaborate scavenging by separate pumps to ensure high mean effective pressures. Lightweight high-speed gasoline engines usually are of the four-stroke type, whereas medium-size gas and oil engines are built to operate on either cycle. This question was also discussed in Sec. 15-7.

Arrangement of Cylinders.—Low-speed engines and engines in which the floor space does not matter are built with horizontal cylinders. Vertical in-line cylinders are preferred for higher speeds. When the engine must be more compact, V-type and radial engines make still shorter and lighter constructions.

Number of Cylinders.—Horizontal engines are built with one, two, or four cylinders, either all parallel, or, in larger sizes, in tandem and double-tandem arrangements to obtain a better utilization of the running gear.

Vertical engines are built from one to sixteen cylinders. Four-, six-, and eight-cylinder arrangements have their advantages in the increase of uniformity of rotation, power, and better balancing. The same applies to five- and seven-cylinder engines, which are not used so often. A greater number of cylinders is used chiefly to increase the power of the engine without increasing its height and width. As will be shown later, the weight of an engine per horsepower decreases with an increased number of cylinders.

In radial engines the number of cylinders used are 3-5-7-9, and 14 and 18 staggered in two banks.

Single and Double Action.—The great majority of engines are built single acting. Double action presents considerable difficulties in designing and construction, particularly in vertical engines. It is used only in very large engines of several hundred horsepower per cylinder, and up.

17-2. Pressures. *Compression Pressure and Ratio.*—As already discussed, in engines with fuel and air mixed before compression, the compression pressure is limited by the danger of preignition and detonation. In compression-ignition engines the pressure of compression must be high enough to ensure ignition at any conditions.

TABLE 17-1.—COMPRESSION, PRESSURES, AND AIR DATA

Engine cycle	Ignition	Fuel	Range of compression		Mean effective pressure p_e psi	Air used average $1 + e$
			Pressure, p_c psig	Compression ratio, $r = V/V_c$		
Explosion, automotive	Electric	Gasoline	90–125	5.2– 7.5	75–100	1.0
Explosion, aircraft	Electric	Gasoline, aviation	100–125	5.7– 7.9	95–135	1.0
Explosion	Electric	Alcohol	130–225	6.0– 9.0	60– 85	1.0
Explosion	Electric	Natural gas	100–130	5.0– 6.0	74– 90	1.3
Explosion	Electric	Coke-oven gas	100–135	5.0– 6.2	70– 90	1.3
Explosion	Electric	Producer gas	110–160	5.4– 7.0	60– 80	1.25
Explosion	Electric	Blast-furnace gas	120–190	5.7– 7.9	60– 82	1.2
Explosion	Hot bulb	Kerosene, fuel oil	50– 75	3.2– 4.2	50– 70	2.0
Combustion, air injection	Compress.	Fuel oil	450–525	14 –16	75– 85	1.8
Combination, airless injection, low speed	Compress.	Fuel oil	370–450	12 –14.5	75– 90	1.75
Combination, high speed	Compress.	Fuel oil	450–650	14 –18	80–110	1.3–1.7

The approximate compression pressures p_c and average compression ratios r are given in Table 17-1. As already explained, a higher compression ratio gives a better thermal efficiency and increases the mean effective pressure, or power output of the engine, but at the same time results in higher stresses in the working parts. In spark-ignition engines a higher compression ratio also requires special precautions in designing the compression chamber to avoid preignition and detonation.

Mean indicated pressure p_i can be computed from the theoretical expression (5-15), (5-17), or (5-19), as the case may be, using instead of k an average value of n as given in Chap. 12-14. The theoretical value thus calculated must be multiplied by the relative efficiency η_r, expression (16-3).

Instead of assuming values of n and n_r, the pressure p_i can be assumed directly, if there is previous experience.

For automobile engines, a fair empirical value of p_i is twenty times the compression ratio r; for modern aircraft engines, p_i is slightly higher, about twenty-two times r.

A third method is based on volumetric relations. The heat value Q^l of one unit of fuel, Table 4-2, divided by the corresponding volume of air-fuel mixture taking into consideration the theoretical amount of air a, cu ft/lb if Q_l is in Btu/lb or cu ft/cu ft if Q_l is in Btu/cu ft, and excess air e and multiplied by 778 represents the work in foot-pounds of 1 cu ft mixture. Now foot-pounds divided by cubic feet give pressure in pounds per square foot and represent the pressure doing the work, which is the mean indicated pressure in question with a 100 per cent efficiency. Actual p_i is found by taking into consideration the different losses, multiplying the pressure by the ideal-cycle efficiency η_{id}, relative efficiency η_r, and charge efficiency η_{ch}, and dividing by 144 to change pounds per square foot to pounds per square inch,

$$p_i = 5.4Q_l\eta_{id}\eta_r\eta_{ch}/[v_f + a(1 + e)] \tag{17-1}$$

where v_f is the volume of 1 lb fuel, Table 4-2, and for heat value Q_l in Btu/cu ft $v_f = 1$.

The charge efficiency should be determined by expression (12-20) or (12-21), as the case may be, making suitable assumptions for pressures and temperatures.

In a two-stroke engine the scavenge efficiency η_{sc} must be used instead of η_{ch}; η_{sc} can be estimated by using the curves of Figs. 15-19 to 15-21.

Brake mean effective pressure, often called simply *mean effective pressure* and designated p_e, is the mean indicated pressure p_i multiplied by the mechanical efficiency η_m

$$p_e = p_i\eta_m \tag{17-2}$$

It is very convenient to use the value p_e when comparing different engines or designing a new engine. Figure 16-7 permits the estimating of η_m. Table 17-1 may give a certain guidance in checking p_e, obtained from expressions (17-1) and (17-2), for four-stroke engines and Table 15-1, for two-stroke engines.

17-3. Engine Speed. *Rotative Speed.*—The selection of the rotative speed, or revolutions per minute, is often governed by the purpose for which the engine is designed or by the speed of the machinery with which the engine must be connected. There is a tendency to increase the speed of engines of all types in order to lower their weight and cost. The speed varies between 100 and 5000 rpm, and no set rules can be given for its selection.

TABLE 17-2.—ENGINES WITH DIFFERENT SPEED CHARACTERISTICS

Hp per cylinder	Bore, in.	Stroke, in.	Rotative speed, rpm	Piston speed c, ft/min	Speed factor c_s	Speed classification
10	4⅞	6	900	900	8.1	*M*edium
70	9	12	900	1800	16.2	*H*igh
475	18	21	900	3150	28.4	*S*uperhigh
10	4⅝	6	1000	1000	10	*H*igh
70	12¼	16	375	1000	3.8	*M*edium
475	32	60	100	1000	1.0	*L*ow

Piston speed in present engines varies from 650 to 3600 fpm, with a tendency toward a further increase in some types. The purpose is the same as with the rotative speed—to decrease the weight of the engine. Several empirical formulas were proposed for determining suitable piston speeds, but the applications of internal-combustion engines are so varied that no general formula is possible.

Speed Factor.—Engines can be classified into several classes as to "speediness": low-, medium-, and high-speed types. However, unless a defined "yardstick" is used, the designations remain vague. The use of either the rotative speed n or the piston speed c as a speed characteristic is unsatisfactory, because they both are inadequate for the purpose.

To illustrate this inadequacy two groups of three engines in each were taken, Table 17-2. One group has the same rotative speed of 900 rpm, but the engines obviously belong to three different classes so far as speediness is concerned, as indicated in the last column. The same applies to the second group of engines having the same piston speed of 1000 fpm.

The reason why rotative speed alone is not suitable as a speed characteristic is that it does not take into consideration the size of the engine. On the other hand, the piston speed is influenced excessively by the piston stroke. A good speed characteristic, commonly called now *speed factor*, is obtained as a product of n and c and for the sake of obtaining small figures divided by 100,000[1]

$$c_s = nc/100{,}000 \tag{17-3}$$

Substituting for the piston speed its expression

$$c = nl/6 \tag{17-4}$$

where l is the length of the stroke in inches, gives

$$c_s = n^2l/600{,}000 \tag{17-5}$$

Engines with the same c_s are in the same class as far as speediness is concerned. For the sake of classification, engines with a speed factor below 3 can be called *low-speed*, with a speed factor between 3 and 9 *medium-speed*, from 9 to 27 *high-speed*, and above 27 *super-high-speed* engines.

Stroke-bore Ratio.—A check of several hundred sets of engine data made by the author has shown that the ratio of stroke to bore l/d varies between 0.9 and 1.9 with only a high-speed opposed-piston two-stroke engine having $l/d = 2.5$. There does not exist any definite relation between l/d and the speediness of an engine. Low-speed engines with a short stroke-bore ratio are about as common as long-stroke high-speed engines.

Fair average values for stroke-bore ratio l/d are 1.0 for air-cooled aircraft engines and 1.2 to 1.35 for all other engines.

17-4. Principle of Similitude.—The designing of an engine is based on former experience, meaning information about the dimensions and behavior of existing engines under conditions more or less approaching those for which the new engine is being designed. It is very helpful to have exact means to interpolate or extrapolate such information correctly.

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[1] MALEEV, V. L., A Proposed Speed Characteristic, *ASME Trans.*, vol. 54, No. 20, 1932; *Power*, vol. 75, p. 512, 1932.

Such means is the use of the principle of similitude—designing the new engine so that it will be similar to existing units operating satisfactorily. The first thing is to determine what constitutes a similar engine. To be complete the similarity must be geometrical, kinetical, and dynamical, which means that the engines must have

(1) the same stroke-bore ratio l/d,
(2) the same speed factor c_s, and
(3) the same mean effective pressure p_e.

These three conditions determine all important features of a new engine which is intended to be similar to an existing one.

As the basic figure it is convenient to take the bore of an engine because the bore determines the majority of other dimensions. Therefore, in the following, all items will be expressed as functions of the bore d.

The above stated three requirements can be developed into the following more detailed conclusions in regard to similar engines:

1. Linear dimensions are proportional to the bore; this follows from the condition l/d constant.

2. Revolutions per minute n are inversely proportional to the square root of the bore as can be seen from expression (17-5) in conjunction with a fixed value for l/d.

3. The piston speed is proportional to the square root of the bore which results from expression (17-4) in conjunction with statement 2.

4. The horsepower per cylinder is proportional to the bore raised to the 2.5 power, being proportional to piston displacement (d^3) and rotative speed ($1/\sqrt{d}$).

5. Gas velocities in the valves and ports are proportional to the square root of the bore, being proportional to piston speed c, statement 3.

6. Friction velocities are proportional to the square root of the bore because they are proportional to d and n, statement 2.

7. Forces produced by gas pressures are proportional to the bore in the second power because they are a function of the piston area.

8. Inertia loads are proportional to the bore in the third power because they are proportional to the weight of reciprocating parts which are proportional to d^3, proportional to the stroke d, and proportional to the second power of the rotative speed $(1/\sqrt{d})^2$.

9. Specific bearing pressures and stresses from acting forces are the same in similar engines.

10. Specific bearing pressures and stresses from inertia loads are proportional to the bore.

11. Elastic deflections from acting forces are proportional to the bore.

12. Elastic deflections from inertia loads are proportional to the bore in the second power.

13. The natural frequency of vibration of any part is inversely proportional to the bore, as can be seen from the general expression

$$f = k\sqrt{F/W} \tag{17-6}$$

where k is a constant, F is the force per inch of deflection (d^2/d), and W is the weight (d^3).

14. The weight of the engine is proportional to the bore in the third power and the number of cylinders.

15. The weight per horsepower is proportional to the square root of the bore.

16. The weight per cubic foot or cubic inch of piston displacement is the same in similar engines.

TABLE 17-3.—INFLUENCE OF BORE IN SIMILAR ENGINES

Constant:	**Stroke/bore ratio l/d**
	Speed characteristic c_s
	Mean effective pressure p_e
	Bearing pressures from acting forces
	Stresses from acting forces
	Weight per cu ft of piston displacement
Proportional to $1/d$:	**Natural frequency of vibration**
Proportional to $1/\sqrt{d}$:	**Rotative speed n, rpm**
Proportional to $\sqrt{d}$:	**Piston speed c**
	Gas velocities
	Friction velocities
	Weights per hp
Proportional to d:	**Linear dimensions**
	Elastic deflections from acting forces
Proportional to d^2:	**Acting forces from gas pressures**
	Elastic deflections from inertia loads
Proportional to $d^{2.5}$:	**Horsepower per cylinder**
Proportional to d^3:	**Inertia loads**
	Weights, lb/cylinder

These conclusions are shown classified in Table 17-3.

This table shows at a glance how to determine a certain dimension, what to expect in certain respects, and helps to design an engine which will in the best possible manner maintain all the desirable features of a similar smaller or larger engine.

17-5. Determination of Main Dimensions.—As explained above, the designing of a new engine starts with the selection of the main characteristics—speed factor c_s or at least approximate classification of the engine in regard to speed; stroke-bore ratio l/d; and mean effective pressure p_e. The last can be decided upon, either by immediate selection based on performance of similar engines, or by selecting values entering in expression (17-1) and the mechanical efficiency. Simultaneously the compression pressure p_c or the compression ratio r must be selected.

The next thing is to decide about the number of cylinders. With these data determined, the piston displacement and consequently the bore and stroke of the engine necessary to develop the required horsepower N are computed using the general expression

$$N = plan/33{,}000 \tag{17-7}$$

This equation can be better adapted by expressing l, a, and n in terms of d, and with the designation

$$l/d = q \tag{17-8}$$

The revolutions are found from expression (17-5),

$$n = \sqrt{600{,}000c_s/qd} = 774.6\sqrt{c_s/qd} \tag{17-9}$$

This gives for a two-stroke single-acting cylinder

$$N = p_e qd \times 0.7854d^2 \times 774.6\sqrt{c_s/qd}/(33{,}000 \times 12)$$

or

$$N = 0.001534p_e d^{2.5}\sqrt{qc_s} \tag{17-10}$$

The diameter of the cylinder is therefore

$$d = [651.5N/(p_e\sqrt{qc_s})]^{0.4} \tag{17-11}$$

For a four-stroke engine the number of working strokes is only half of n giving

$$N = 0.000767p_e d^{2.5}\sqrt{qc_s} \tag{17-12}$$

and

$$d = [1303N/(p_e\sqrt{qc_s})]^{0.4} \tag{17-13}$$

If the rotative speed n is given or assumed instead of the speed factor c_s, then expression (17-10) can be presented as

$$N = np_e qd^3/505{,}000 \tag{17-14}$$

and

$$d = 79.6 \sqrt[3]{N/np_e q} \tag{17-15}$$

For four-stroke engines the value of n in equations (17-14) and (17-15) must be divided by 2.

If an engine is coupled to an a-c generator, then the speed found by expression (17-9) usually must be adjusted, either lowered or raised, to obtain the prescribed number of cycles or frequency f, according to the relation

$$n = 60f/p \tag{17-16}$$

where p is the number of pair of poles. This will change somewhat the speed factor c_s, and the bore should be found by expression (17-15) rather than by (17-11) or (17-13).

After the bore, stroke, and speed are found, a theoretical indicator diagram must be drawn and the dimensions of the different parts determined, taking into consideration the various acting forces such as gas pressures, pressures acting upon the piston, forces of inertia, and other resistances and heat stresses.

Example 17-1.—Determine the bore and stroke of a four-stroke single-acting engine of 200 bhp operating on natural gas which has the following analysis: 5 per cent H_2, 75 per cent CH_4, 15 per cent C_2H_6, and 5 per cent N_2.

Selected: low-speed type with a speed factor $c_s = 2$, stroke-bore ratio $l/d = q = 1.4$, number of cylinders, 2.

The heat value of 1 cu ft gas is, using data in Table 4-2,

$$274 \times 0.05 + 910 \times 0.75 + 1633 \times 0.15 = 1065 \text{ Btu/cu ft}$$

Theoretical air-fuel ratio from Table 4-2

$$a = 2.39 \times 0.05 + 9.52 \times 0.75 + 16.65 \times 0.15 = 9.76$$

Select in accordance with Table 3-1, the actual air $(1 + e) = 1.35$.

Select from Table 17-1, an average compression ratio $r = 5.5$ which with a suction pressure $p_a = 13.7$ psi, Table 13-1, and with $n = 1.35$ gives a compression pressure,

$$p_c = 13.7 \times 5.5^{1.35} = 137 \text{ psia or } 122.3 \text{ psig}$$

Figure 5-10 gives an efficiency $\eta_{t,d} = 0.365$. The relative efficiency can be estimated conservatively as $\eta_r = 0.85$.

The charge efficiency is computed from expression (12-20), assuming a total increase of the charge temperature of about 110 F, which gives

$$T_d = 60 + 110 + 460 = 630 \text{ R}$$

also $t_e = 1000$ F, the pressure of the exhaust gases $p_e = 15.7$ psia, and with the main gas constituent being CH_4, enough accurately $n_1/n_2 = 1$:

$$\eta_{ch} = \frac{520(5.5 \times 13.7/630 - 15.7/1460)}{14.7 \times (5.5 - 1)} = 0.857$$

With these data, using equation (17-1),

$$p_i = 5.4 \times 1065 \times 0.365 \times 0.85 \times 0.857/(1 + 9.76 \times 1.35) = 107.5 \quad \text{psi}$$

Estimating the mechanical efficiency as 78 per cent, Fig. 16-7, by expression (17-2)

$$p_e = 107.5 \times 0.78 = 83 \text{ psi}$$

The horsepower N per cylinder is $200/2 = 100$ which should be increased for overload by about 10 per cent, giving $N = 110$ hp.

Now from expression (17-13) the cylinder bore

$$d = (1303 \times 110/83 \sqrt{1.4 \times 2})^{0.4} = 16.0 \text{ in.}$$
$$l = 16 \times 1.4 = 22.4 \text{ or } 22\tfrac{1}{2} \text{ in.}$$

By equation (17-9)

$$n = 774.6 \sqrt{2/22.5} = 231 \text{ rpm}$$

Example 17-2.—Determine the main dimensions of a two-stroke single-acting airless-injection 500-bhp oil engine using fuel oil which has an elementary analysis of 85 per cent carbon and 12 per cent hydrogen. The engine must be direct connected to an electrical generator with a speed of 360 rpm.

Select number of cylinders, 4; stroke-bore ratio, $q = 1.25$.

The heat value of the oil is, using expression (4-41),

$$Q_h = 14{,}520 \times 0.85 + 61{,}045 \times 0.12 - 1150 = 18{,}517 \text{ Btu/lb}$$

The lower heat value Q_l, is from expression (4-40),

$$Q_l = 18{,}517 - 9441 \times 0.12 = 17{,}385 \text{ Btu/lb}$$

Selecting a compression ratio $r = 13.5$ and an excess of air $e = 0.6$ gives from Fig. 5-11 an efficiency $\eta_{id} = 0.435$. The relative efficiency may be estimated as $\eta_r = 0.86$. The amount of theoretical air and fuel can be taken according to Table 4-2 as 190.6 cu ft/lb.

The scavenge efficiency of a two-stroke engine depends upon the scavenging process, shape of ports, and excess of air delivered by the pump. Assuming loop scavenging, a scavenge factor $K_{sc} = 1.4$, and a scavenge pressure of 3 psi, from Fig. 15-20, $\eta_{se} = 0.77$.

With these data, using equation (17-1),

$$p_i = \frac{5.4 \times 17{,}385 \times 0.435 \times 0.86 \times 0.77}{190.6 \times 1.60} = 88.5 \text{ psi}$$

With a mechanical efficiency of about 0.75 the brake mean effective pres sure becomes

$$p_e = 88.5 \times 0.75 = 66.4 \text{ psi}$$

The horsepower per cylinder is 500/4 = 125. Adding 10 per cent of overload gives 137.5 hp.

Using the above found values in expression (17-15) gives for a two-stroke engine:

$$d = 79.6 \sqrt[3]{137.5/360 \times 66.4 \times 1.25} = 13.17 \text{ or } 13\tfrac{1}{4} \text{ in.}$$
$$l = 13.25 \times 1.25 = 16.54 \text{ or } 16\tfrac{1}{2} \text{ in.}$$
$$c_s = 360^2 \times 16.5/600{,}000 = 3.6$$

This is a medium-speed engine according to the definition.

17-6. Problems.—**1.** Compute the theoretical brake mean effective pressure for an aircraft engine using octane as fuel; no excess air, 6.5 compression ratio. Estimate the relative and mechanical efficiencies; state reasons for the selected values.

2. Compute the theoretical brake mean effective pressure for a six-cylinder automobile engine using 65° API gasoline; assume a small excess of air, a compression ratio of 6.25. State reasons for all assumptions.

3. Find the piston speed of an engine with a 48-in. stroke and a rated speed of 120 rpm.

4. Determine the speed factor c_s and the speed classification of the engine in problem 3.

5. Determine the speed factor c_s and the speed classification of a 9-in. × 11-in. engine at 1000 rpm.

6. Determine the piston speed, speed factor, and speed classification of a 10-in. × 12-in. engine running at 514 rpm.

7. An existing four-cylinder four-stroke oil engine, with a 12-in. bore, 15-in. stroke, 320 rpm, develops 220 hp and weighs 30,000 lb. Using the principle of similitude, find bore, stroke, revolutions per minute, and weight of a similar engine that will develop 280 hp.

8. An existing four-cylinder four-stroke 8-in. × 10-in. gas engine develops 120 bhp at 600 rpm and weighs 7250 lb. Using the principle of similitude, find the bore, stroke, speed, and weight of a similar five-cylinder engine that will develop 250 bhp.

9. Determine the bore and stroke of a two-stroke double-acting single-cylinder 1000-hp engine operating on blast-furnace gas that has the following analysis: H_2, 3 per cent; CH_4, 1; CO, 27; CO_2, 10; balance, N_2. The engine is to be direct-connected to a blower making 90 double strokes per minute.

10. Determine the bore, stroke, and revolutions per minute of the engine discussed in problem 1 to develop 40 hp per cylinder and to be classified as being on the dividing point between high and super-high speed.

11. Determine the bore, stroke, and speed factor of the engine referred to in problem 2 to develop 85 bhp at 3600 rpm.

12. Determine the main dimensions of a four-stroke single-acting six-cylinder airless-injection 420-bhp oil engine. The fuel oil to be used contains 86.0 per cent carbon, 12.5 per cent hydrogen, and 1 per cent sulphur.

The speed factor is to be about 3, near the dividing point between low and medium speed.

13. Determine the main dimensions and speed of a four-stroke single-acting four-cylinder airless-injection 260-bhp oil engine. The fuel oil to be used has a 32° API gravity. The engine will be coupled to a 60-cycle a-c generator and should have a speed factor placing it as an average medium-speed engine.

14. Determine the main dimensions and speed of a natural-gas engine using gas of average composition. The engine is of the four-stroke single-acting type and must develop 250 bhp in two cylinders. A speed factor of about 4 is desirable.

15. Determine the main dimensions of a four-stroke compression-ignition oil engine with a speed factor $c_s = 24$, to develop 120 bhp.

16. Determine the main dimensions of a two-stroke compression-ignition oil engine with a speed factor $c_s = 24$, to develop 120 bhp, using a uniflow scavenging scheme with one piston per cylinder.

CHAPTER 18

FOUR-STROKE INTAKE AND EXHAUST

18-1. Timing.—Instead of the theoretical opening and closing of the intake and exhaust valves on dead centers and staying open during 180° of crank travel, the valves of four-stroke engines open before and close after the respective dead centers. The object of this early opening and late closing is to increase the average valve lift, and to reduce the gas velocities and therefore their flow resistance in the intake and exhaust ports. The opening of the intake valve occurs only slightly before dead center, but the closing is always retarded considerably to make use of the high velocity of the gases induced in the intake pipe and thus to obtain a greater charge into the cylinder.

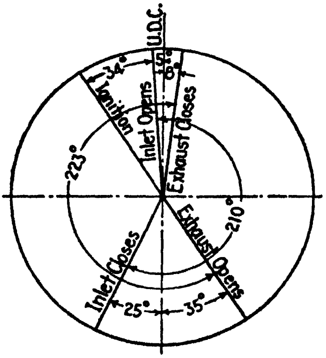

FIG. 18-1.—Timing of a low-speed spark-ignition engine.

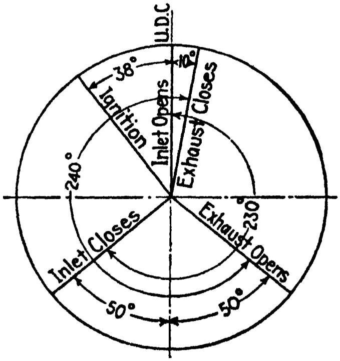

FIG. 18-2.—Timing of a high-speed spark-ignition engine.

The conditions are reversed for the exhaust valve: It can close near the dead center but must open considerably before the other dead center to reduce the pressure in the cylinder to as near the atmospheric pressure as possible before the exhaust stroke starts.

The proper timing is a function of the engine speed. With increase of the engine speed the intake valve is closed later and the exhaust opened earlier. Typical timing diagrams are: Fig. 18-1 for a low-speed gas engine, Fig. 18-2 for a high-speed automobile engine, Fig. 18-3 for a medium-speed compression-ignition oil engines, and Fig. 18-4 for a high-speed compression-ignition oil engine.

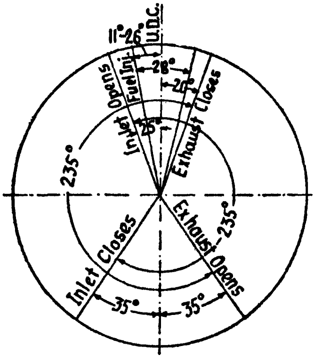

FIG. 18-3.—Timing of a medium-speed oil engine.

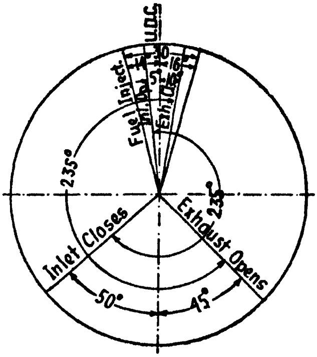

FIG. 18-4.—Timing of a high-speed oil engine.

The actual points of opening and closing in degrees of crank travel are not as important as the amount of valve opening, which governs the difference between the cylinder pressure and the atmospheric pressure. As stated in Sec. 12-2, it is more important to keep the intake pressure rather than the exhaust pressure close to the atmospheric pressure.

TABLE 18-1.—RANGE OF VALVE TIMING, FOUR-STROKE ENGINES
(Degrees of crank travel before (*b*) or after (*a*) dead center)

Type of engine	Intake			Exhaust		
	Opens	Closes	Duration, deg	Opens	Closes	Duration, deg
Aircraft	35*b*–10*a*	35–90*a*	215–305	80–42*b*	0–45*a*	233–305
Automotive	15*b*–15*a*	35–70*a*	215–261	65–38*b*	5–30*a*	225–285
Motorcycle	28 –20*b*	50–62*a*	245–285	65–50*b*	10–37*a*	240–280
Gas, stationary	20*b*–10*a*	20–40*a*	200–240	55–30*b*	5–20*a*	210–240
Oil, air injection	30*b*– 0	5–40*a*	210–243	50–30*b*	0–25*a*	210–240
Oil, airless injection	20 – 8*b*	20–40*a*	210–230	45–20*b*	5–25*a*	210–240
Oil, airless injection, high speed	20 –10*b*	30–55*a*	225–245	50–35*b*	10–25*a*	225–250

In addition to the consideration of speed, the best valve timing can be determined only by actual tests, as it depends greatly on the design of the intake and exhaust passages. The ranges of valve timing and durations of valve openings as encountered in engines of proved performance are given in Table 18-1, which shows a very great variation in all types of engines. Comparison of the Figs. 18-3 and 18-4 shows also that the duration of overlap of the intake and exhaust does not depend upon the engine speed.

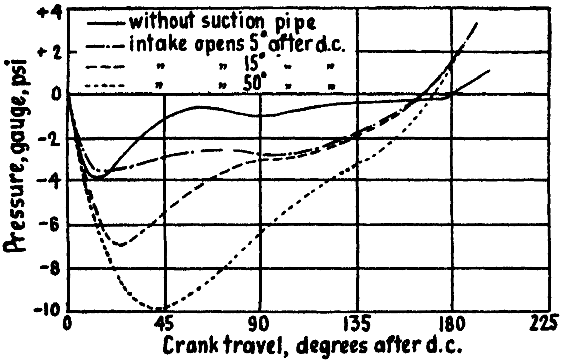

FIG. 18-5.—Effect of intake-valve opening on cylinder pressure.

The opening of the intake valve after the dead center, used in some engines, mostly high-speed, Table 18-1, has for its object an increase of the charge efficiency due to a pulsating pressure. Figure 18-5 shows the increase of pressure at the beginning of the compression, if the intake has a suction pipe and the intake valve opening is retarded,[1] as compared with the pressure without a suction pipe.

In double-acting engines the timing is kept the same for both ends, giving slightly different piston positions on account of the connecting-rod angularity.

18-2. Valves. *Types.*—Most four-stroke engines are equipped with so-called *poppet* or *mushroom valves.* The other type used is the *slide valve* found in a few automobile engines. Ports controlled by piston edges of two-stroke engines actually act as large slide valves.

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[1] LIST, H., Die Strömung in Saugrohren von Verbrennungskraftmaschinen, *Z. Ver. deut. Ing.* vol. 85, No. 13, p. 304, 1941.

Flow of Gases.—In dealing with internal-combustion engines the gas velocities usually are expressed in feet per minute. For a small pressure difference the gas head h in the fundamental equation (7-1) of flow can be expressed as in formula (7-3)

$$h = (p_1 - p_2)w_w/0.4332w_g \qquad (18\text{-}1)$$

where p_1 is the gas pressure causing the flow, psi,
p_2 is the pressure against which the gas flows, psi,
w_w is the density of water, 62.37 lb/cu ft at 60 F,
w_g is the density of the gas, lb/cu ft at the pressure p_2.

Thus the gas velocity in feet per minute can be expressed

$$v = 5780\varphi\sqrt{(p_1 - p_2)/w_g} \qquad (18\text{-}2)$$

where φ is a coefficient of velocity and varies from 0.85 to 0.95.

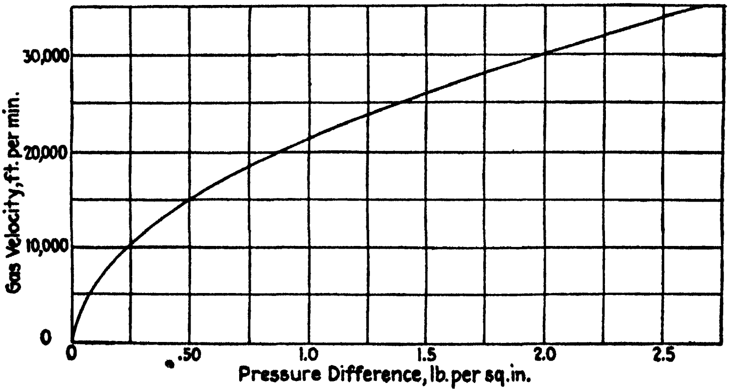

FIG. 18-6.—Influence of pressure difference on gas velocity.

Theoretical velocities, computed by expression (18-2), with $\varphi = 1$, for standard air, $w_g = 0.0763$, and for pressure differences occurring in engines, give a curve, Fig. 18-6, which shows that even a small pressure difference is sufficient to produce a very high velocity.

The general formula for gas velocity, valid for any pressure difference, $p_1 - p_2$, for an adiabatic flow is

$$v = \sqrt{\frac{2gkRT_1}{k-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]} \qquad (18\text{-}3)$$

With $k = 1.41$, $g = 32.2$, introducing the velocity coefficient φ, and changing from fps to fpm, equation (18-3), with equations

(2-9) and (2-52) becomes

$$v = 13{,}430\varphi \sqrt{(T_1 - T_2)c_p} \tag{18-4}$$

where c_p is the specific heat; for air at low temperatures one can use $c_p = 0.24$.

Critical Pressure.—If the back pressure $p_2 < 0.53p_1$, the gases in the throat, valve, or port will expand only to the critical pressure $p_c = 0.53p_1$, and this pressure or corresponding temperature should be used in determining v by expression (18-3).

Coefficient of Discharge.—If a is the average port area in square feet, then the volume of gas flowing into or out of the engine cylinder may be expressed in cubic feet per minute,

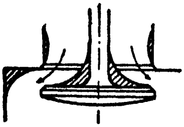

FIG. 18-7.—Streamlining of the intake.

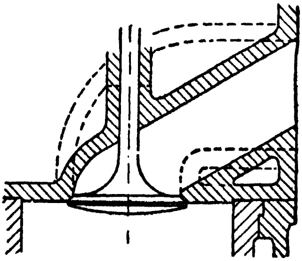

FIG. 18-8.—Streamlining of the exhaust.

$$V = \alpha a v \tag{18-5}$$

where α is the *coefficient of contraction* which can be determined according to the rules of hydraulics. For approximate computations, α may be taken between 0.65 and 0.85, the lower value for ports which have sharp edges and higher values when the edges are rounded. The product of the two coefficients is called the *coefficient of discharge* $\mu = \alpha\varphi$.

For poppet valves three factors influence the value of μ:

1. The form and length of the passage of the valve.
2. The size of the valve.
3. The relative lift of the valve.

Rounding of all edges of the valve and seat, particularly on the upstream side of an intake value, Fig. 18-7, raises the value of μ.[1]

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[1] WOOD, G. B., JR., HUNTER, D. U., TAYLOR, E. S., and TAYLOR, C. F., Air Flow through Intake Valves, *Trans. SAE*, vol. 50, No. 6, pp. 213, 252, June, 1942.

A valve located directly over the piston has a higher μ than one located on the side in a pocket. The value of μ increases with the diameter d of a valve. For a flat seat the ratio h/d, where h is the lift, acts in an opposite direction; for a small lift, $h = 0.1d$, the coefficient μ may reach 0.80, gradually decreasing with the increase of h to about 0.62 for $h = 0.25d$.[1] For a conical seat the best value of μ is attained at about $h = 0.2d$.[2]

Gas Velocities.—It is customary to base the valve-opening data upon the mean gas velocity v_g, which is computed as a function of the mean piston speed c,

$$v_g = c(A/a) = c(D/d)^2 \tag{18-6}$$

where A is the piston area,
a is the valve area,
D and d are the respective diameters.

The piston speed varies from 0 to maximum $c_{\max}$ and back to 0 at the end of each stroke. The ratio $c_{\max}/c = \pi/2 = 1.57$. Thus the maximum gas velocity $v_{g\max}$ through a valve will be 1.57 times greater than v_g found by expression (18-6).

The greater the gas velocity v_g, the greater becomes the pressure difference $p_1 - p_2$ in expression (18-2) or (18-3) necessary to maintain the flow. In order to ensure a high charge efficiency of the cylinder, both the back pressure at the end of the exhaust stroke and the drop of pressure at the end of the intake should be small. Good practice is to obtain mean gas velocities v_g as computed from expression (18-6), between 7,000 and 10,000 fpm for stationary engines and from 10,000 to 15,000 fpm for automotive and aircraft engines. However, the pressure difference is influenced not only by the velocity but by turbulences as well. Thus, changing the exhaust connection in a cylinder head, Fig. 18-8, from the dotted-line shape, reduced the back pressure considerably in spite of an increase in the gas velocity, because the straighter outlet reduced the forming of turbulences.

Expression (18-6) does not take into consideration either the pressure p or the temperature t of the gas and thus gives fictitious values for v_g, useful only for comparison or as a design characteristic. A more correct absolute value of v' can be found by using the characteristic equation, charge efficiency η_{ch}, and timing data; thus,

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[1] DENNISON, KUCHLER, and SMITH, Experiments on the Flow of Air through Engine Valves, *Trans. ASME*, O. & G. Power, vol. 53, p. 60, 1931.

[2] WOOD, HUNTER, TAYLOR, and TAYLOR, *op. cit.*, p. 215.

$$v_g' = 14.7 v_g T \eta_{ch}\ 180/[520p\ (180 + \alpha + \beta)] \qquad (18\text{-}7)$$

where α is the number of degrees of crank travel from the valve opening to dead center and β is the number of degrees from the next dead center to the closing of the valve.

When the mean gas velocities are computed by expression (18-7), they should not exceed 12,000 fpm for the intake and 18,000 fpm for the exhaust for stationary engines and about 50 per cent higher in automotive engines at maximum speed. Since the valves are being opened very gradually and the full opening approximately corresponds to the maximum piston speed, the pressure difference causing the flow should be computed not for the mean velocity but for the maximum velocity.

Example 18-1.—Find the mean and the corrected velocities for the intake and exhaust in a 10-in. × 18-in. engine running at 260 rpm. The diameter of both valves is 2¾ in. The temperature of the incoming charge is 75 F, the intake pressure is 0.6 psi below atmospheric, the exhaust temperature is 900 F, the back pressure is 1.2 psi above atmospheric, the charge efficiency is 0.87. Both the intake and exhaust are open through 212°.

The nominal mean velocity, by expression (18-6), is

$$v_g = 2 \times 18 \times 260 \times 10^2/(2.75)^2 \times 12 = 10{,}320 \text{ fpm}$$

The corrected intake velocity is, since $T = 75 + 460 = 535$ R,

$$v_i' = \frac{10{,}320 \times 535 \times 14.7 \times 0.87 \times 180}{520 \times (14.7 - 0.6) \times 212} = 8170 \text{ fpm}$$

The corrected exhaust velocity is

$$v_e' = \frac{10{,}320 \times (900 + 460) \times 14.7 \times 0.87 \times 180}{520 \times (14.7 + 1.2) \times 212} = 18{,}400 \text{ fpm}$$

Example 18-2.—For the engine in the previous example, determine the theoretical pressure difference during the intake and exhaust.

Assuming for the intake a velocity coefficient $\varphi = 0.85$ and for the exhaust $\varphi = 0.93$ the theoretical pressure difference during intake may be found from expression (18-2) with $w_g = w_1 = 14.1 \times 144/53.35 \times 535 = 0.0711$ lb/cu ft

$$p_1 - p_2 = [8170 \times 1.57/(5780 \times 0.85)]^2 \times 0.0711 = 0.484 \text{ psi}$$

The exhaust back pressure may be calculated with the exhaust pressure in the manifold as $14.7 + 0.5 = 15.2$ psia and

$$w_2 = 15.2 \times 144/53.35 \times 1360 = 0.041 \text{ lb/cu ft}$$
$$p_1 - p_2 = [18{,}400 \times 1.57/(5780 \times 0.93)]^2 \times 0.041 = 1.19 \text{ psi}$$

which is very close to the data given in Example 18-1. The greater differences found by measurement are due to turbulences.

Valve Seats and Lifts.—Valve seats usually are conical with an angle $\alpha = 45°$, Fig. 18-9. In order to obtain a greater opening area for a limited valve lift, some inlet valves are made with a flatter angle, usually 30°. For exhaust valves an angle smaller than 45° makes the edges of the heads too thin and more subject to the corrosive action of the exhaust gases. Flat sets, $\alpha = 0°$, are used very seldom and only on horizontal valves, as foreign particles may lodge on a flat seat of a vertical valve.

The ring-shaped valve opening depends upon the perpendicular distance between the seat surfaces $h \cos\alpha$, and its area may be expressed approximately as

$$a = \pi d_1 h \cos\alpha \qquad (18\text{-}8)$$

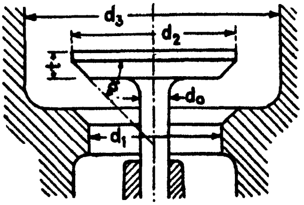

FIG. 18-9.—Main valve dimensions.

Neglecting the stem diameter, the area of the circular opening of the valve port is

$$a_c = \pi d_1^2/4 \qquad (18\text{-}9)$$

The amount of lift which gives the same opening as the circular valve port is obtained by equating expressions (18-8) and (18-9)

$$h = 0.25 d_1/\cos\alpha \qquad (18\text{-}10)$$

Expression (18-10) shows that with a conical seat the lift must be higher than with a flat seat. Under the above condition for a flat seat, $h = 0.25d_1$; with $\alpha = 30°$, the lift $h = 0.29d_1$; and for $\alpha = 45°$, the lift $h = 0.35d_1$. In actual engines the lift is made smaller, between $0.12d_1$ and $0.23d_1$, in order to reduce the hammering of the valve against the seat when closing. The main advantage of a conical seat is its self-centering tendency.

If a valve is located in a pocket or recess, which is rather common practice in horizontal and L-head vertical engines, the diameter of the recess d_3, Fig. 18-9, must be such as not to restrict the passage around the valve disk, or

$$0.7854(d_3^2 - d_2^2) \geq 0.7854 d_1^2$$

which gives

$$d_3 \geq \sqrt{d_1^2 + d_2^2} \qquad (18\text{-}11)$$

Double Valves.—In high-speed engines even the largest possible valves may give very high gas velocities and great pressure differences. By using two valves for the intake and two for the exhaust, each valve having a slightly greater diameter than $0.71d$ of the single valve, a greater opening area and lower gas velocities are obtained. Another advantage is a smaller lift h, if the ratio h/d is kept the same as for a single valve. Other advantages are a lower valve weight, which reduces the forces to which the valve mechanism is subjected, and better cooling through an increase of the valve-seat area, if the same seat width is used.

18-3. Design of Valves. *Materials.*—Valve cages are made of hard, close-grained cast iron; separate valve seats are made of the same cast iron as the cages, or of special heat-resisting bronze. Valves are made of nickel steel; exhaust valves of high-speed engines are made of special heat-resisting alloys, as chrome-nickel steel, and for severe service, as in aircraft engines, the contact cone is surfaced by welding on a stellite layer. Valves of larger stationary gas and oil engines and all water-cooled valves are made with cast-iron disks and medium-carbon-steel stems.

Dimensions.—The thickness t of the valve disk can be determined using the formula for a flat plate[1]

$$t = k_1 d_1 \sqrt{p/S} \tag{18-12}$$

For cast-iron disks, $k_1 = 0.54$ and the allowable stress S may be taken as 4000 psi; for steel $k_1 = 0.42$ and S for carbon steel 8000 psi, and for high-grade alloy steel up to 15,000 psi. To decrease the weight the thickness t may be decreased toward the edges to $0.65t$.

The projected width of the seat, $b = 0.5(d_2 - d_1)$, may be determined from the condition that the bearing stress due to the explosion pressure p_{max} should not exceed a safe value S_b. This gives

$$b = 0.5d_1[\sqrt{S_b/(S_b - p_{max})} - 1] \tag{18-13}$$

where S_b can be taken as 4000 psi. The width b may be determined also from an empirical formula[2]

$$b = 0.05d_1 \text{ to } 0.07d_1 \tag{18-14}$$

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[1] MALEEV, V. L., "Machine Design," p. 252, International Textbook Company, Scranton, Pa., 1939.

[2] LISTON, T., "Aircraft Engine Design," p. 198, McGraw-Hill Book Company Inc., New York, 1942.

The diameter of the valve stem can be made approximately

$$d_0 = d_1/8 + \tfrac{3}{16}\text{ in.} \qquad (18\text{-}15)$$

The compressive stress in the stem of the exhaust valve should be checked when the valve is opened, and the cylinder pressure at this moment, as taken from the indicator diagram, may reach 50 to 60 psi.

A heavy stem decreases the wear of the guide and facilitates the conduction of heat from the exhaust valve to the cooling water.

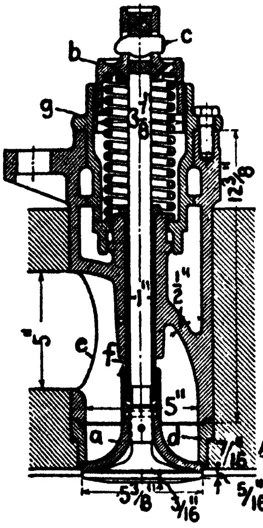

Fig. 18-10.—Exhaust valve.

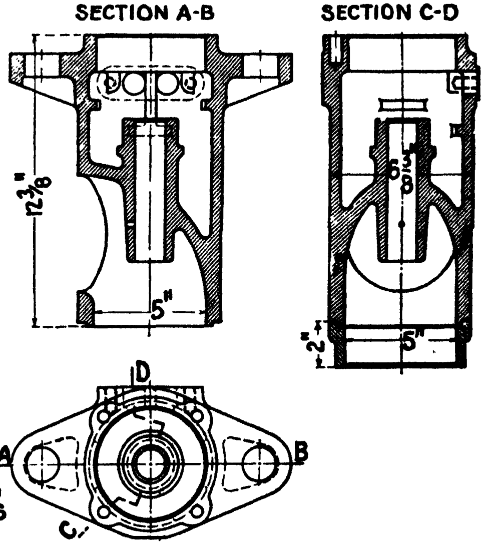

Fig. 18-11.—Exhaust valve cage.

The length of the guide does not depend directly on the size of the valve but more on the valve mechanism and must be increased if the side movement of the actuating levers is considerable.

Constructions.—An automobile valve is an example of a forged all-steel intake valve, whereas Fig. 6-3 shows a typical intake valve with a cast-iron disk and a steel stem. Figure 6-4 shows a typical valve and cage assembly also using a cast-iron valve and, with the exception of the gas admission valve *g*, is very similar to air intake valves of medium- and large-size oil engines.

Exhaust valves of small engines are made interchangeable with intake valves. In aircraft engines on account of the exceedingly high temperatures the exhaust valves are made of heat-resistant steels. Larger valves are made with water-cooled cages. Figure 18-10 shows the details of an exhaust valve for medium- and large-size low-speed oil engines. The disk has a cast-iron washer *a* screwed on and riveted to the stem with an extension serving as protection for the valve stem and guide against hot gases. The upper spring seat and guide *b* is held by a key *c* and has a recess and holes for lubricating the valve stem.

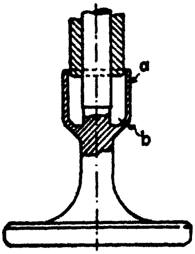

FIG. 18-12.—Exhaust valve with deflector.

The valve cage is made up of four pieces: the valve seat *d* cast of special close-grained iron; the main piece *e*, with the exchangeable cast-iron bushing *f*; and the upper piece *g* closing the water space. The main part *e* is shown separately in Fig. 18-11, which shows the openings for the admission and discharge of the cooling water and the ribs which, with corresponding ribs on piece *g*, act as guides for better water circulation.

Figure 18-12 shows an all-steel exhaust valve of a medium-size oil engine. The valve has a gas deflector *a*, which prevents the hot exhaust gases from penetrating into the clearance between the stem and guide, burning the lubricating oil there, and causing the valve to stick. The hole *b* prevents lubricating oil from accumulating in the cup and forming a residue, which would interfere with a proper seating of the valve.

Figure 18-13 shows a hollow aircraft-engine exhaust valve partially filled with sodium. Under the influence of the high temperature the sodium melts and moves up and down with each opening and closing of the valve, and thus helps to transfer the heat to the outside and prevents the valve from overheating.

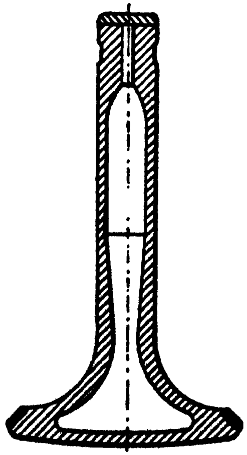

FIG. 18-13.—Sodium-cooled exhaust valve.

In order to prevent the inserting of an intake valve instead of an exhaust valve, in some engines the same sodium-cooled valves are used for both exhaust and intake.

Exhaust-valve cages should be symmetrical about a plane including the center line of the valve and normal to the exhaust port. This prevents warping of the cage by the hot exhaust gases which results in leaky and burned valves. In water-cooled valve cages the question of symmetry is sometimes neglected in favor of shaping the exhaust port so as to reduce the gas-discharge resistance, such as shown in Figs. 18-10 and 18-11.

Spring Retainers.—The upper spring seat, also called the *spring retainer*, in low-speed engines is of cast iron, Fig. 18-10. In high-speed engines it is a steel washer shaped to fit the top of the spring and attached to the top of the valve stem by some type of removable fastener. The type widely used, both in gasoline and oil engines, is provided with a conical recess in which the valve stem is locked by means of a conical split collar, called the *keeper*, Fig. 18-14, which fits around the stem and into grooves turned in it. The pressure of the spring on the retainer holds the keeper tightly in place; yet it can be easily removed by depressing the spring while holding the valve in its closed position.

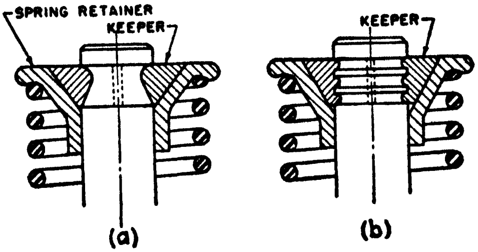

FIG. 18-14.—Valve-spring retainers.

Some large engines have valve-spring retainers screwed on the stem end and locked by a cap lock nut with a hardened upper surface.

18-4. Valve Gear. *Cams.*—Most valves are operated by means of cams. The cam contour is formed by two arcs joined by two flanks, which may be straight lines or curves tangent to both arcs, Fig. 18-15.

The cam imparts a motion to the valve by means of a cam follower, of which there are three main types:

1. The roller follower, Fig. 18-15*a*.
2. The mushroom follower, Fig. 18-15*b*.
3. The pivoted follower, Fig. 18-15*c*.

Various combinations of cams and followers are also used, one of them being the pivot follower and roller made into a rocker arm, Fig. 18-16, and used in most horizontal and some vertical engines.

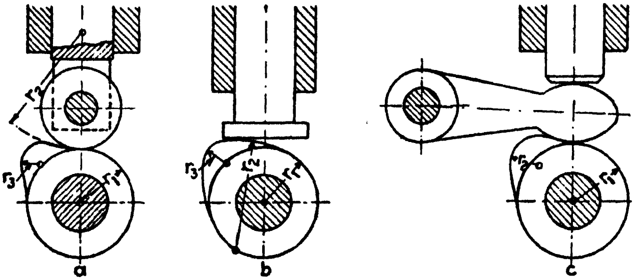

Fig. 18-15.—Types of cams and cam followers.

Lift.—The motion of the valve and its lift depends upon the design of both the cam and the cam follower. A valve-lift diagram is plotted after the actual valve lifts at small crank-angle increments are found. An example is shown in Fig. 18-17 for a tangential cam and roller follower. The clearance *s* between the base circle of the cam and the roller results in a lag *o-a*.

The curve of valve velocity is plotted next, velocity v being the first derivative of lift x with respect to the time, or crank angle, $v = dx/dt$.

The second derivative is the acceleration $a = d^2x/dt^2$, which may increase, remain constant, or decrease up to the point where the flank merges into the cam nose. The sudden change of acceleration from a positive value *cd* to a negative value *ce* at this point may cause the roller to leave the cam and, when returning under the action of the spring, to hammer the cam. Therefore it is desirable to shape the cam so as to decrease the fluctuation of acceleration, and to obtain an acceleration curve similar to that shown in dotted lines $h - i$.

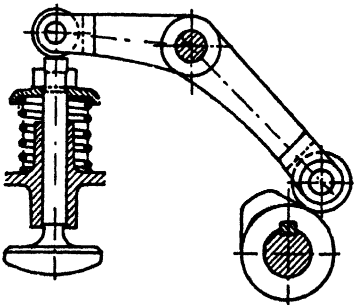

Fig. 18-16.—Cam and rocker arm mechanism.

Roller and mushroom followers give diagrams of lifts, velocity, and acceleration, which are symmetrical with regard to the opening and closing of the valve, whereas the pivoted follower gives unsymmetrical diagrams.

Sometimes it is desirable to keep the valve wide open, *i.e.*, in its maximum-lift position, for a certain period of time. This is achieved by making the cam nose concentric with the base radius, as shown in the example, Fig. 18-16. During the period of *dwell*, the velocity and acceleration of the valve are zero. In drawing the diagrams for valves with rocker arms the leverage ratio must be taken into account.

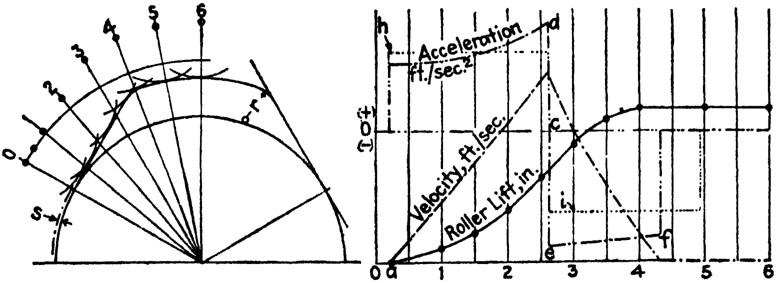

FIG. 18-17.—Tangential cam and its lift-velocity-acceleration diagram.

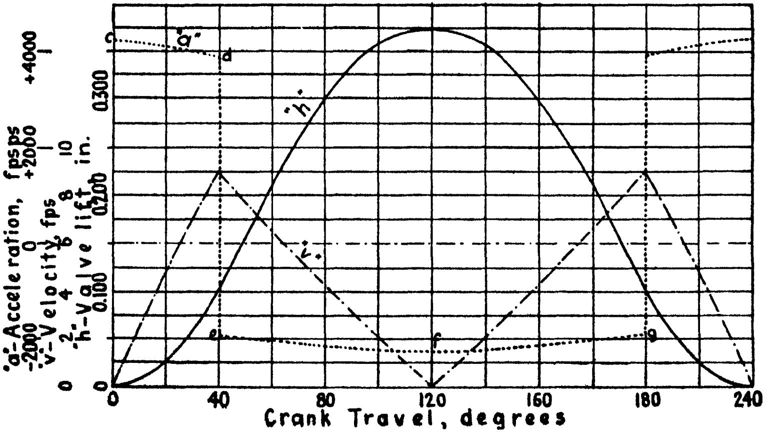

FIG. 18-18.—Curved flank cam with flat-mushroom follower—engine speed 3000 rpm.

Figure 18-18 gives the lift, velocity, and acceleration curves for a cam that has curved flanks, no dwell, and a mushroom follower. The valve diameter is 1¾ in. and the lift is ⅜ in.; the rated speed of the 4¼-in. × 5¼-in. engine is 3000 rpm.

Clearance and Valve Lash.—The links of the mechanism from the cam to the valve are provided with a small clearance, from 0.004 to 0.030 in., depending upon the arrangement and engine size, to ensure seating of the valve under any condition. A change of the clearance varies the time of opening and closing of the valve. The clearance creates a lost motion, often referred to as *valve lash.* Because of the valve lash the cam follower acquires a certain velocity when it touches the valve stem; therefore valve velocity must increase instantaneously from zero to cam-follower velocity. This requires considerable acceleration and results in a shock action. The shock produces a noisy action and wear. In high-speed engines this shock may be eliminated by dividing the opening cam flank into two parts. The first part, after effecting the necessary clearance lift and acceleration for the follower, allows it to come to rest at this position; and the second part produces the desired valve motion, starting with a zero velocity of the whole mechanism.

The temperature effect on the valve gear causes different elongations of the various parts. This has a tendency to change the clearance between the cam and the cam follower. Such a change affects the engine timing and must be taken into account by the designer. In an engine in operation the required clearance must be checked and adjusted when the engine is hot. In most engines the adjustment is made possible by an adjustable screw and lock nut at one end of the valve rocker arm. The clearance is measured directly with a feeler gage inserted between the valve-stem tip and the rocker-arm roller or hardened tappet.

Automatic valve-lash adjusters are used to avoid the necessity of a clearance between the cam and follower, regardless of whether the engine is cold or warm. The adjusters give a constant contact between the cam and follower and eliminate shock action at the beginning of the valve opening. They also eliminate the need for manual adjustment to take care of wear at various points of the valve gear. At present two types of automatic adjusters are in use, the mechanical and hydraulic.

Mechanical Adjuster.—A mechanical valve-lash adjuster usually consists of a special cam on the rocker-arm end, which acts as a tappet upon the valve-stem tip. The cam *a* is slightly eccentric, Fig. 18-19, and a spiral spring *s*, located in the hollow fulcrum pin *b*, has a tendency to turn it, as shown by the arrow, when the valve is closed, and thus takes up the clearances between the various parts of the valve-actuating gear.

Hydraulic Adjuster.—Hydraulic lash adjusters, sometimes called *hydraulic lifters*, may be built into either the cam followers or the tappets striking the valve stem. With valve-in-head engines they are generally built into the end of the rocker arms or valve bridges, which act directly on the ends of the valve stems. An example is shown in Fig. 18-20: A small cylinder containing a

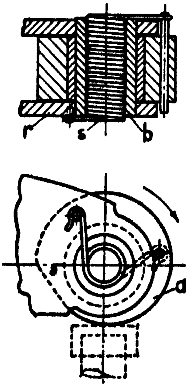

FIG. 18-19.—Valve lash take-up. (*Courtesy of General Motors Corporation.*)

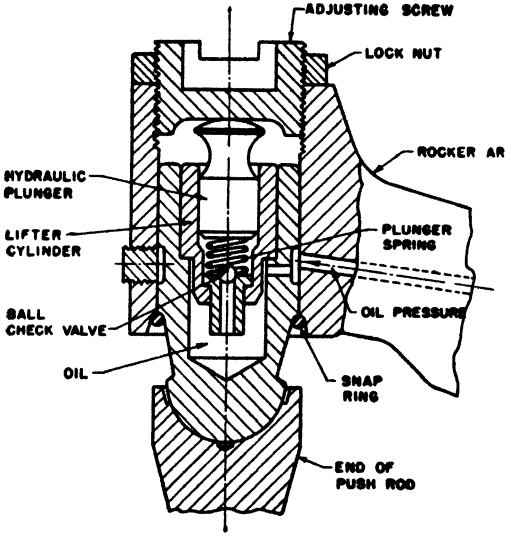

FIG. 18-20.—Hydraulic lash adjuster.

piston or plunger, a spring, and a ball check valve is interposed between the poppet valve and its lifting mechanism. In operation, oil under pressure from the lubricating oil system enters the small cylinder past the ball check valve and is trapped therein above the plunger. Any force exerted against the outer end of the plunger will be transmitted to the cylinder, mounted in the valve gear, by the entrapped oil. The valve is thus actuated just as though the valve lash were taken up mechanically. Since the spring inside the cylinder acts to force the plunger outward, any clearance between the valve and its lifter will be taken up, and the oil pressure will immediately fill up the cylinder through the check valve. If the valve stem expands, there is sufficient leakage of oil past the plunger to permit it to move in slowly so that the valve will not be prevented from seating.

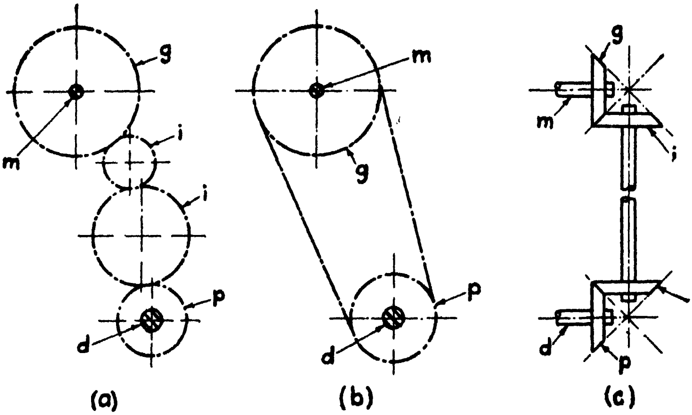

FIG. 18-21.—Types of camshaft drives.

Camshafts.—In small engines the cams and camshafts are machined from one piece. In larger engines the cams are machined separately, hardened, and keyed on a camshaft made of turned and ground steel. The camshaft diameter must be large enough to ensure torsional and longitudinal rigidity and should be supported by bearings with large surfaces in order to reduce the wear to a minimum.

In horizontal engines the camshaft is driven from the crankshaft by means of bevel or helical gears. In small vertical engines the camshaft drive is effected by spur gears or a silent chain, Fig. 18-21*a* and *b*, where *d* is the crankshaft and *m* the camshaft. In larger vertical engines the camshaft is driven by a train of spur or helical gears, Fig. 18-21*c*.

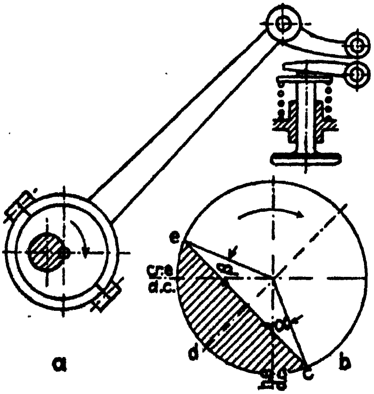

FIG. 18-22.—Eccentric mechanism.

Eccentric Mechanism.—On account of the very large forces necessary to operate valves of large horizontal gas engines, they are often equipped with eccentrics, controlling wiping cams and swinging cam followers, Fig. 18-22*a*. The valve diagram, Fig. 18-22*b*, shows that only part of the eccentricity is used for opening of the valve, from point *c* to *d*. The closing of the valve *d-e* is effected by a spring identical with those of cam-operated valves. The advantage of this construction is a smooth operation, without noise, and with very small wear.

18-5. Springs. *Force.*—The acceleration curve, Fig. 18-17, shows that from point *e* to point *f* the acceleration is negative; the follower tends to leave the cam. During this time a force must be applied to the valve mechanism to cause all its links to remain in contact with each other. In small- and medium-size engines this force is obtained from the valve spring and is a function of the mass of the parts moved by the cam

$$F = ma = (W/g)a \qquad (18\text{-}16)$$

where F is the force, lb,

W is the weight of the moving parts, lb,

g is the acceleration of gravity, 386 in.psps,

a is the acceleration of the moving parts, in.psps, taken from the acceleration diagram.

In larger engines the accelerations of the valve-gear parts must be taken care of by a positive kinematic chain.

In overhead valves the spring must also support their weight. The friction in the stem guides can usually be neglected.

In engines with throttling of the intake the spring must be strong enough to prevent the opening of the exhaust valve by the vacuum created in the engine cylinder during the suction stroke. This vacuum may reach 7 to 8 psi.

The valve springs are usually helical, cylindrical springs and have a straight-line relationship between force and deflection. In order to avoid breaking of the spring by repeated loading it is advisable to select a spring with a greater number of coils, 10 to 14. This ensures a small difference of tension between the positions where the valve is closed and fully open.

Deflection and Strength of Springs.—The formulas for the deflection and the safe load of cylindrical springs made of round steel wire are

$$y = 8iD^3F/d^4G \qquad (18\text{-}17)$$

and

$$F = \pi d^3 S_s / 8Dk \tag{18-18}$$

where y is the deflection, in.,
i is the number of free coils,
D is the mean diameter of coils, in.,
F is the safe load, lb,
d is the wire diameter, in.,
G is the modulus of rigidity, for steel about 11,400,000 psi,
S_s is the allowable shear stress, psi,
k is a stress-increase coefficient.

For heavy, closely coiled springs, as used for valves,

$$k = (4c - 1)/(4c - 4) + 0.615/c \tag{18-19}$$

where $c = D/d$ and should be $c \geqq 5$.

The allowable stress may be taken for $d = 0.5$ in. as 68,000 psi for SAE 1095 and up to 88,000 psi for SAE 6150 steel. For wires of other diameters the allowable stress must be multiplied by the size factor,[1]

$$e_{ss} = 0.86 + 0.07/d \tag{18-20}$$

Double Springs.—If the force that must be exerted by the spring is very large, the spring may require a very heavy stock and a large D. Using two concentric springs will make each smaller and fit them better in the available space. Concentric springs are also used in aircraft engines for the sake of safety; if one breaks, the other will continue to operate the valve.

Spring Surge.—If the natural frequency of vibration of a spring is equal, or is in a simple ratio to the frequency of the disturbing force, there will be synchronous vibration and the center coils will begin to surge back and forth, changing the force exerted by the spring. This phenomenon, and how to overcome it, is discussed in Chap. 25.

18-6. Sleeve Valves.—Of the various attempts to replace the poppet valve the sleeve valve is the only type that has had a certain success. The best known is the Knight double-sleeve valve which was used in several American and still is being used in some European automobile engines. Its main advantage is the absence of valve noise and freedom from wear if properly lubricated;

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[1] *Inco*, vol. 9, No. 1, p. 21, 1931.

another advantage is that it permits the use of the best possible form of combustion chamber. The disadvantages are of a mechanical nature.

The valve consists of two concentric sleeves, Fig. 18-23, sliding between the piston and the cylinder wall. The sleeves are driven by cranks of an auxiliary shaft operated at half the engine speed. The inner sleeve reaches the dead center slightly after the piston reaches the upper dead center, and the outer sleeve lags about 80° of the auxiliary-shaft travel behind the inner sleeve. Slots in the sleeves and in the cylinder wall form the intake and exhaust ports.

The timing of the valves can be investigated by valve diagrams in which the movement of the slot in one sleeve is superimposed upon the movement of the slot in the other sleeve and both are drawn with relation to the slot in the cylinder wall. The valve opening is represented by an area where all three slots

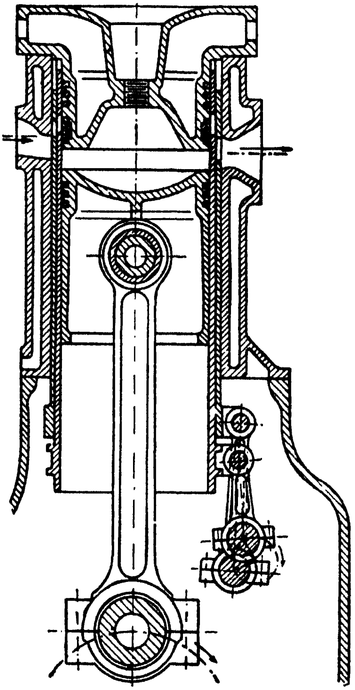

FIG. 18-23.—Double-sleeve valve engine.

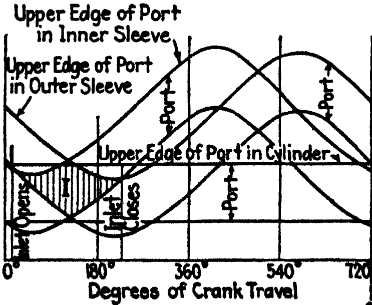

FIG. 18-24.—Valve diagram for intake.

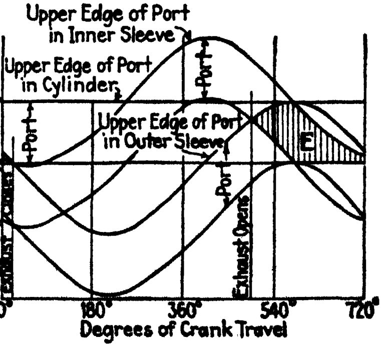

FIG. 18-25.—Valve diagram for exhaust.

register, area *I* for the intake, Fig. 18-24, or area *E* for the exhaust, Fig. 18-25. These diagrams show the moments of opening and closing and the width of opening at any crank angle.

Lately the interest in the sleeve valves has been revived in view of certain advantages which they present for high-speed compression-ignition engines.

18-7. Manifolds. *Intake.*—The best cross section is a circle, as it gives the least resistance to the charge flow. The limits of charge velocities and the corresponding manifold sizes were discussed in Sec. 7-5. To this may be added that, with downdraft carburetors in automobile engines, the charge velocity in the riser can be reduced to 3000 fpm at 1000 rpm.[1] This low velocity reduces the pressure drop and increases the charge efficiency. However, it is considered desirable, with both downdraft and updraft manifolds, to have horizontal distributor sections.

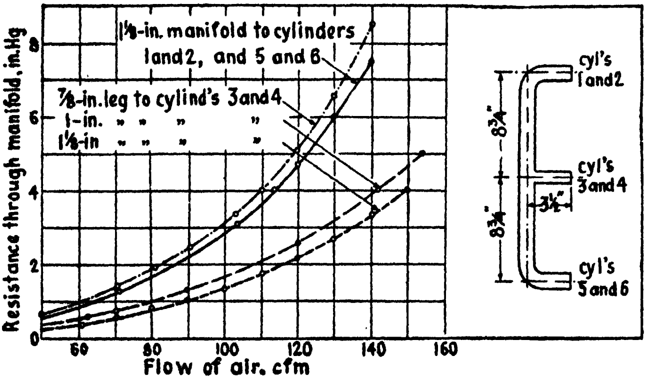

FIG. 18-26.—Resistance to air flow through a three-port intake manifold.

In order to obtain the maximum output from a multicylinder engine, the charge distribution to all cylinders must be equal. This in turn requires equal resistances between the carburetor and the intake port to each cylinder. Since the distances to various ports are different, it seems advisable to equalize the resistances by changing the cross section. Figure 18-26 shows such an equilization scheme and at the same time gives information as to the amount of vacuum which is created in the manifold.[2] Some designers believe, however, that the inertia of the longer gas column offsets the greater resistance and makes the change of sections unnecessary.

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[1] LICHTY, L. C., "Internal-Combustion Engines," 5th ed., p. 271, McGraw-Hill Book Company, Inc., New York, 1939.

[2] TAUB, A., Mixture Distribution, *SAE Journal*, vol. 26, p. 454, 1930.

In the case of carburetor engines the problem of designing the intake manifold is complicated by the fact that the droplets of liquid gasoline, suspended in the charge stream, have a tendency to be deflected to one side of the pipe whenever there is a change of velocity or direction of flow, as in the case of the throttle action. The junction of the vertical stack to the horizontal distributing pipes presents such problems. For an even distribution of the fuel charge to both horizontal branches it is very important to have the horizontal pipe normal to the center line of the vertical pipe, Fig. 18-27*a*. The streamlining of a T-connection, with the object of reducing the resistance to flow, is of doubtful value.

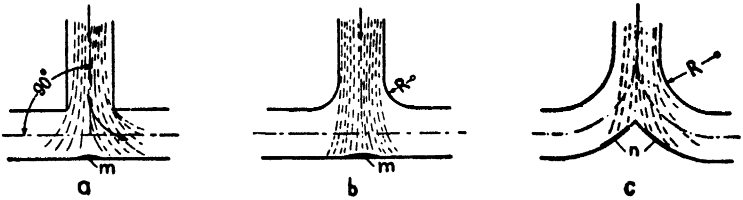

Fig. 18-27.—Typical intake-manifold T-connections.

Streamlining that results in a local velocity reduction tends to precipitate some of the liquid particles at *m*, Fig. 18-27*b*. Full streamlining, Fig. 18-27*c*, will give an uneven fuel distribution by the V-shaped projection, because the fuel is never distributed quite uniformly in the vertical pipe. Also there is a tendency to give a liquid deposit at *n*. Similar conditions exist for an updraft connection. However, streamlining of the intake manifold gives good results with sections through which the mixture flows in one direction only.

Exhaust. *Back Pressure.*—As pointed out before, Sec. 12-2, each psi of back pressure reduces the mep about 1 per cent because of the extra work done by the piston during the exhaust stroke. The dilution caused by the exhaust gases left in the cylinder due to an increased back pressure lowers the charge efficiency by another 0.5 per cent.[1] Thus for 1 psi increase of the back pressure the power output decreases about 1.5 per cent.

Exhaust Overlapping.—According to Table 18-1 the average duration of the exhaust-valve opening is about 220°. This means that with a four-cylinder engine the valve opening of each cylinder will overlap $(220 - 720/4)/2 = 20°$ on each end. Since the valve lift is small at the beginning of opening and toward closing, there will not be any appreciable interference. With a six-cylinder engine the overlapping will be

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[1] Lichty, *op. cit.*, p. 286.

$$(220 - 720/6)/2 = 50°$$

on each side, which must increase to a certain extent the resistance created by the simultaneous discharge of gases from two cylinders, although the valves are not opened very much. With eight cylinders in line, the overlapping becomes

$$(220 - 720/8)/2 = 65°$$

and to prevent an increase of the back pressure, if a single manifold is used, its diameter must be increased, some 10 to 20 per cent, depending upon the actual timing and valve lifts.

Types of Manifolds.—The best exhaust system has separate leads from each cylinder, streamlined at some distance from the cylinder back into one or several exhaust pipes. Such a system is used for racing cars, in which the maximum power output is the main object.

The next best arrangement is to divide the exhaust manifold into so many branches that no two cylinders exhaust into the same branch simultaneously. For a four-cylinder engine this would mean two branches, one for each two cylinders, which is hardly necessary. For a six-cylinder engine two branches will be sufficient. To one branch will be connected the ports of the first three cylinders whose firing occurs 240° part, and to the second one those of the last three cylinders. The two branches may be streamlined into a common exhaust line.

The exhaust manifold of an eight-cylinder engine should be divided into two sections, each corresponding to a four-cylinder-engine manifold with a minimum of overlapping. This is easily done with a V-eight engine and is more difficult with an 8-in-line engine. The simplest solution is to have individual stacks from each cylinder, streamlined as much as possible into a common exhaust header.

In all cases the exhaust manifold should have sufficient cross sections with restrictions to flow reduced as much as possible by streamlining and the use of large-radius bends.

Figure 18-28 shows the exhaust manifold for a four-cylinder automobile engine with streamlined connections from each port to the common header; *h* is a connection for heating the intake manifold.

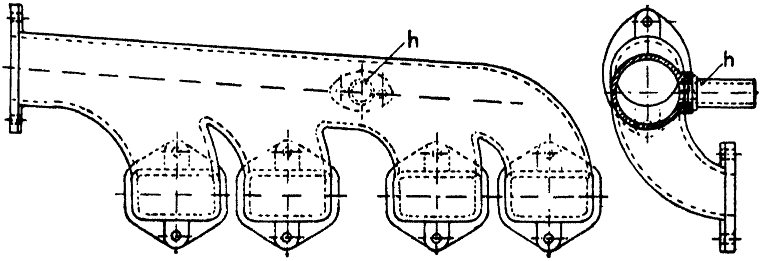

FIG. 18-28.—Streamlined exhaust manifold.

Heat Expansion.—The temperatures of exhaust manifolds of vertical multicylinder engines vary, depending upon the engine load. The manifold should be fastened to the cylinder heads by flexible connections to permit free expansion and contraction. In larger engines the exhaust manifold usually has a water jacket. The latter should have openings for thorough cleaning, as scale deposits decrease the heat transfer, raise the temperature of the inside wall, and thus cause local strains which may result in cracking of the water jacket.

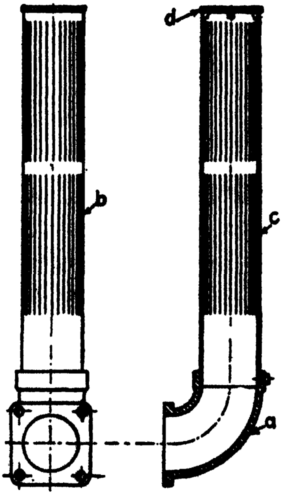

FIG. 18-29.—Intake silencer.

18-8. Accessories. *Intake Silencers.* The air rushing through the intake valve with a high velocity produces a disagreeable hissing noise, which, in large engines, can be heard over a great distance. A simple silencer that at the same time protects the intake mechanism from foreign matter, such as rags or nuts, is made of a piece of thin-wall brass or aluminum pipe *b*, Fig. 18-29, with milled slots *c* not over $\frac{1}{32}$ in. wide and covered with a removable cap *d*; the cast-iron elbow *a* is bolted to the cylinder head. To be effective as a silencer the slots should cease within 12 to 14 in. of the point where the elbow joins the cylinder head. To avoid throttling, the total area through the slots should be 1.5 to 2 times the area of the pipe, which in turn is slightly larger than the intake-valve area.

Air Filters.—The wear of the pistons, rings, cylinders, and valves is materially decreased if the air intake is equipped with a filter. There are on the market a great number of filters of different types suitable for this purpose. For larger engines the air filter is installed just outside the engine building and connected to the engine by a duct made of fairly heavy sheet iron. To prevent throttling of the charge and collapsing of the duct by the atmospheric pressure, if an excessive vacuum is created inside the duct, its cross-sectional area must be at least equal to the total area of all intake valves, even if only one or two cylinders are drawing the charge simultaneously. The air velocity in a duct should not exceed 2500 fpm. With an air filter, there is no intake noise.

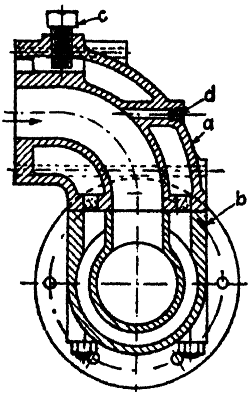

Fig. 18-30.—Water-cooled exhaust elbow.

Exhaust.—The exhaust manifold of vertical oil and gas engines is usually water-jacketed. Its weight and cost are reduced by using a welded construction. The cooling of the exhaust gases protects the engine room from excessive heat and at the same time lowers the back pressure. The exhaust manifold should be connected to the exhaust ports in the cylinder head by elbows to facilitate the removal of the cylinder heads. The elbows also should be equipped with a water jacket, Fig. 18-30. The water spaces of the elbow *a* and manifold *b* are interconnected, and the flow of water can be regulated by a screw *c* opening or restricting the water passage from the cylinder head. The tapped hole *d* is for a pet cock whereby the color of the exhaust can be observed. A better scheme is to use this opening for a thermocouple so that the exhaust temperatures can be observed and the fuel injection adjusted accordingly.

Mufflers.—The exhaust gases escape with a high velocity and produce a disturbing barking sound as they leave the exhaust pipe. The muffling of this sound may be accomplished by different methods—by allowing the gases to expand, by changing the direction of their flow, or by cooling them with injected water. The muffling of the exhaust of multicylinder heavy-duty oil engines usually does not present great difficulties. A cast-iron pot, concrete pit, or welded-steel tank buried in the ground and forming a part of the exhaust line may solve the problem if it has a capacity of about six times the piston displacement of one of the engine cylinders.

Since the exhaust gases of gas engines have a considerably higher temperature, they produce a much louder sound, and its effective muffling often presents difficulty. Figure 18-31 shows an efficient muffler welded of sheet iron. Baffle plates *b* leave openings that force the gases to change their direction of flow many times and that allow the gases to expand gradually. The tangential location of the inlet pipe *c* prevents the gases from impinging on the opposite wall, which would give a ringing metallic sound.

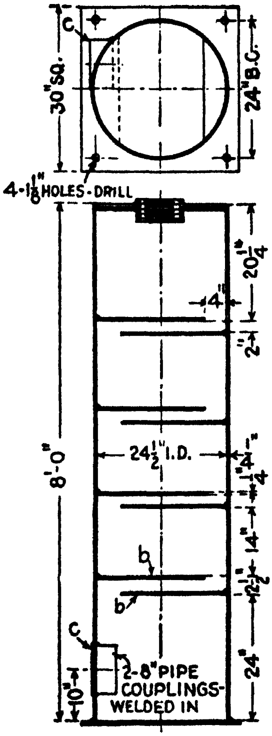

FIG. 18-31.—Welded muffler for a 200-hp engine.

An additional arrangement for reducing the exhaust noise is to have the end of the exhaust pipe or the outlet from a muffler cut on a bias, Fig. 18-32. As the gas flow is gradually diverted from a straight-line motion by the opening on the side, it gradually loses whatever velocity is left in it and thus becomes noiseless.

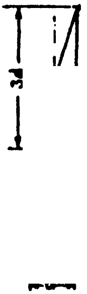

FIG. 18-32.—Exhaust nipple.

There are several makes of exhaust mufflers on the market. The manufacturers call their products *silencers*, but the words muffler and silencer have the same meaning. One commercial muffler is shown in Fig. 18-33: the first, high-pressure high-velocity portion of gases partly expands in the chamber *a*, which thus snubs the velocity, and hence the trade name, partly rushes through into the chamber *b*; from the chamber *c* the gases again partly expand into chamber *d* and partly ooze out directly to the outlet. There is no forcible reversion of the direction of the gas flow, and hence the back pressure is small. This arrangement is very efficient for silencing the exhaust noise and is used on both four- and two-stroke engines.

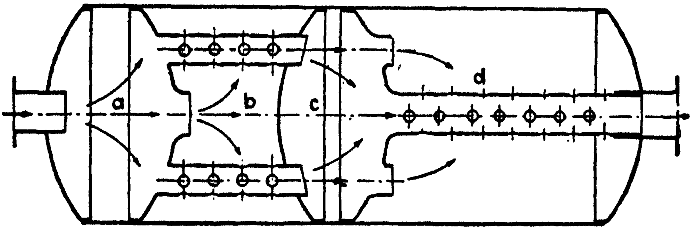

FIG. 18-33.—Double-chamber exhaust muffler.

18-9. Problems.—1. Find the flow of air, in cubic feet per minute, through an opening 4 in. in diameter from an air chamber, where the pressure is 14.95 psia, to the atmosphere.

2. Find the discharge of air, in pounds per minute, through an opening 6 in. in diameter if the air pressure before the opening is 20 psig and after it is atmospheric. Assume two cases: (*a*) sharp edges and (*b*) well-rounded edges of the opening.

3. Find the air velocities created under the conditions: (*a*) of problem 1 and (*b*) of problem 2.

4. (*a*) Calculate the mean gas velocities through the intake valve for an engine with a piston displacement of 60 cu in./cylinder running at 2400 rpm, assuming the valve diameter to bore ratio as 0.4 for six different stroke-bore ratios, 0.5, 1.0 . . . 3.0; (*b*) plot the gas velocities against stroke-bore ratios.

5. Work problem 4 using instead of a constant speed a constant speed factor $c_s = 45$.

6. Work problem 4 for an engine with a piston displacement of 850 cu in. per cylinder running at 514 rpm.

7. Work problem 6 using a constant speed factor $c_s = 5$ instead of the constant rotative speed.

8. Determine the mean and maximum gas velocities in the intake and exhaust valves of a 13-in. × 22-in. four-stroke gas engine running at 240 rpm. The valves are 4½ in. in diameter; 45° conical seat; the maximum lift is ¾ in.; intake opens 10° before, closes 35° after dead center; exhaust opens 40° before, closes 10° after dead center. The charge efficiency is 86 per cent; the suction pressure is 0.95 psi below atmospheric; the temperature of the charge is 180 F; the exhaust temperature is 1050 F; the back pressure is 1.4 psig.

9. Determine the theoretical pressure differences during the intake and exhaust for the engine of problem 8.

10. Determine the main dimensions of a valve for a 14-in. × 24-in., 210-rpm gas engine; timing as in problem 8.

11. Determine the main dimensions and the lift of the intake and exhaust valves for a $5\frac{1}{4}$ in. × 6-in. engine running at 2100 rpm; assume the duration of opening equal to the average value for aircraft engines, Table 18-1.

12. Work problem 11 assuming that two valves are used for the intake and one for the exhaust.

13. Draw a valve-lift diagram with constant accelerations. The engine speed is 1600 rpm; positive acceleration acts 30 per cent of the time; no clearance between cam and valve gear; the cam is symmetrical, and there is no dwell period.

14. In problem 13 the total duration of the cam action is 220° of the crank travel, the valve lift is 0.32 in., and the combined weight of the valve and of the upper spring seat is 10 oz and that of the valve spring is 9 oz. Find: (*a*) the maximum velocity of the valve motion, (*b*) the values of positive and negative accelerations, and (*c*) the maximum force that must exist to keep the roller follower on the cam.

15. Design a valve spring for the data of problem 14.

16. (*a*) Calculate and plot for every 30° of crank travel the theoretical gas velocities during the intake stroke for the $4\frac{1}{4}$-in. × $5\frac{1}{4}$-in. engine which uses the cam of Fig. 18-18; the intake valve begins to open 15° before dead center; (*b*) plot the theoretical vacuum due to these velocities against piston stroke; assume for the connecting rod $L/R = 4.25$; valve diameter $d_1 = 1.5$ in.

17. Design a valve spring for a $3\frac{1}{4}$-in. valve with a lift of $\frac{3}{4}$ in.; the weight of the valve is $2\frac{1}{4}$ lb. The desirable spring forces with the valve closed and open are 140 lb and 240 lb, respectively.

18. Design a valve spring for the intake valve referred to in problem 16 and Fig. 18-18; the valve weighs 12 oz.

19. Assuming that the exhaust valve of the engine referred to in problems 16 and 18 has the same dimensions and weight, design a valve spring for it; the highest vacuum during the intake stroke is 11 in. Hg.

CHAPTER 19

SUPERCHARGING

19-1. General Remarks. *Objects.*—The admittance into the cylinder of an air charge larger than what the cylinder would obtain as the result of the regular suction stroke is called *supercharging*. Supercharging is done by outside means, in contrast to the suction, which in this case is called *natural aspiration*. The increase of the air charge by weight is obtained by increasing the pressure of the air admitted to the cylinder. Sometimes supercharging is called *boosting*.

Supercharging has for its main object the increasing of the air charge to permit the burning of a larger fuel charge and thus to increase the power output of the engine. Supercharging is used in various four-stroke engines in the following main cases:

(*a*) to overcome the effect of high altitude, such as in aircraft engines or in stationary installations in the mountains;

(*b*) to reduce the weight of the engine per horsepower developed, such as in aircraft or racing-car engines;

(*c*) to reduce the bulk of the engine, to fit it into a limited space, such as in locomotive or marine engines;

(*d*) to increase the power of an existing engine when a greater power demand occurs, such as in pipe-line pumping stations.

Thus it can be seen that supercharging has a wide and varied field of applications, and its use is steadily increasing.

19-2. Methods.—The increased air pressure is obtained by using blowers, either of the positive-displacement type, such as Roots blowers, Fig. 19-1*a*, or vane blowers, Fig. 19-1*b*, or of the centrifugal, or turbo, type, Fig. 19-1*c*. Reciprocating-piston blowers are seldom used because they are bulkier, more expensive, and less dependable than blowers of the rotary type.

Roots blowers are used with two or three lobes, with cylindrical or helical surfaces. The three lobes and the helical surfaces are used to obtain a more uniform nonpulsating flow of air. The blowers are usually driven by the engine itself by means of spur, helical, or herringbone gears, silent chains, or V-belts at a speed of two to three times that of the engine. Sometimes, particularly in case (*d*), the blower is driven by an electric motor.

Figure 19-2 shows a three-lobe cylindrical Roots-type supercharger for a 290-hp rail-car engine. The blower has skew-shaped inlet and outlet ports, which give a gradual change of velocity and thus eliminate the noise present in most cylindrical blowers.

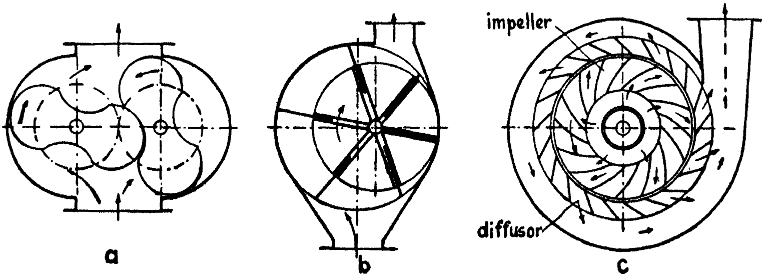

FIG. 19-1.—Types of superchargers.

The centrifugal blowers are used on both low-speed heavy-duty and high-speed, lightweight engines. They are driven either by gears from the engine or directly by an impulse exhaust-gas turbine. The speed of the centrifugal blowers must be comparatively high, around 10,000 to 15,000 rpm for low-speed engines and 15,000 to 30,000 rpm for high-speed engines, such as aircraft engines. If driven by the engine, the speed step-up must be considerable, 1:5 to 1:11.

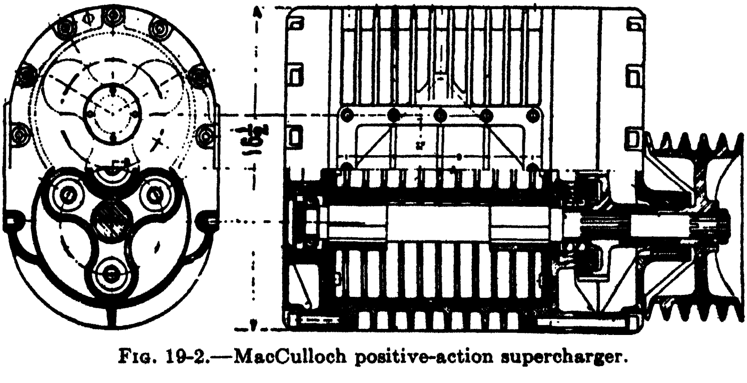

FIG. 19-2.—MacCulloch positive-action supercharger.

Figure 19-3 shows a section through a horizontal centrifugal blower with a spiral discharge header, designed for a mechanical drive. A section through a vertical Buchi turboblower driven by an exhaust-gas turbine is shown in Fig. 19-4. The kinetic energy of the air when it leaves the blower wheel is transformed into pressure by the vanes of a diffuser, Fig. 19-1c or Fig. 19-4.

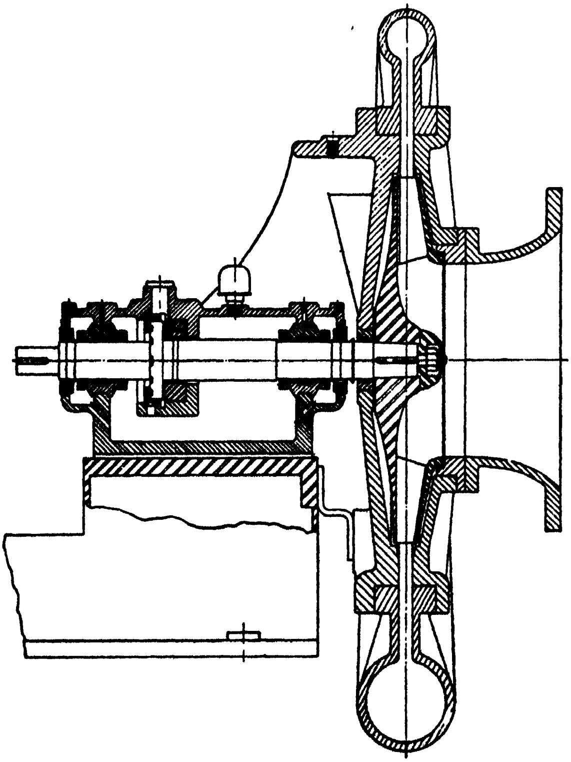

FIG. 19-3.—Centrifugal supercharger.

In order to prevent overlapping of the exhaust pressure fluctuations, separate manifolds from groups of two or three cylinders, which do not exhaust simultaneously, are connected to separate groups of nozzles in the turbine. Figure 19-5 shows a typical turbine-driven supercharger installation in a six-cylinder oil engine. The exhausts from cylinders 1, 2, and 3 discharge into one manifold, and those from cylinders 4, 5, and 6 into the other, lower manifold.

The Buchi turbochargers are built for supercharging four-stroke engines of 250 hp and upward.

Field of Application.—The positive-displacement blowers are used for low- and medium-speed engines, such as in stationary and marine installations. Their output per revolution is very little affected by speed; however, if the engine is reversed, the blower must be driven from an outside source, usually an electric motor. Positive-displacement blowers operate at moderate speeds, not over 4000 rpm, and therefore are somewhat bulky and heavy.

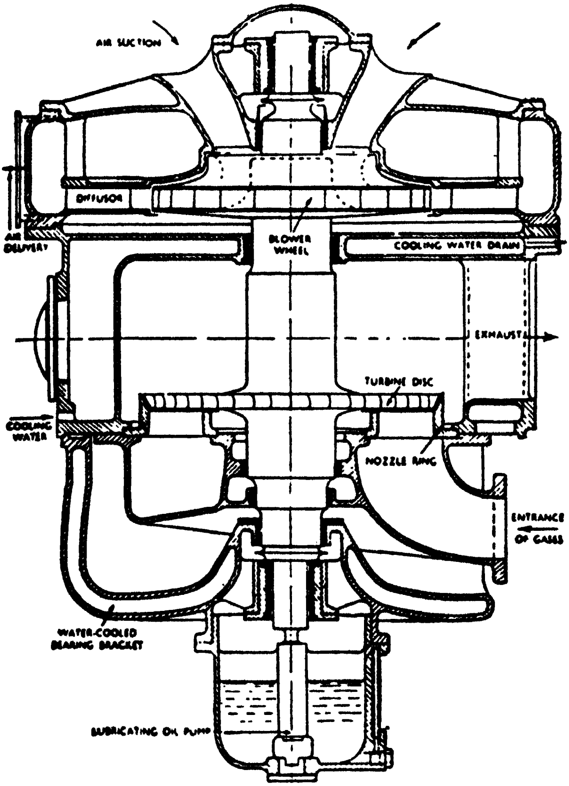

Fig. 19-4.—Centrifugal supercharger driven by an exhaust-gas turbine.

Centrifugal blowers driven by exhaust-gas turbines are small and light and are used for stationary, locomotive, marine, and aircraft engines. With this supercharger an engine can be reversed without affecting the performance of the supercharger.

At reduced speeds, down to 50 per cent or less, the engine torque can be maintained constant without an excessive temperature increase.

Aircraft engines are very good prospects for turbine-driven superchargers. At present most American aircraft engines are using gear-driven centrifugal superchargers. However, aircraft-engine manufacturers are working on the further development of the turbine-driven supercharger, and its wider application can be expected in the near future.

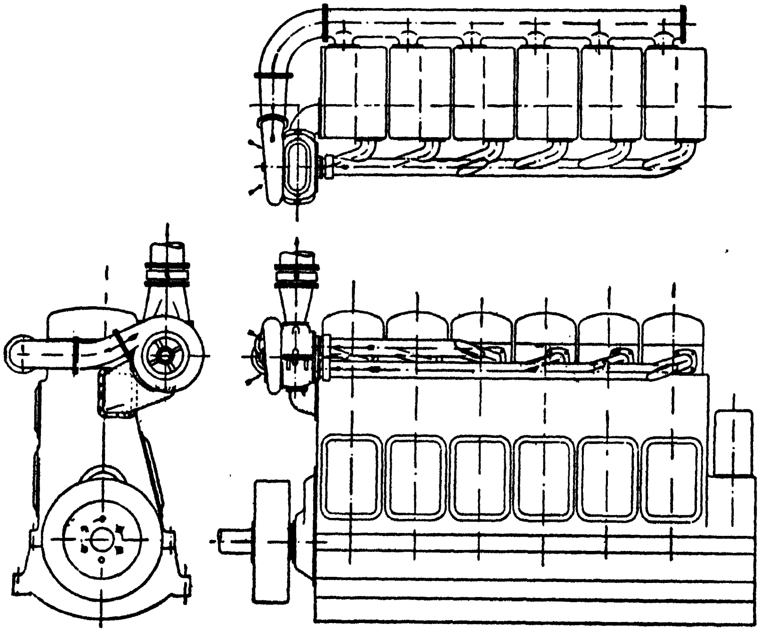

FIG. 19-5.—Turbine-driven supercharger installation.

19-3. Results. *Power Increase.*—An increase of the mean effective pressure by 30 to 50 per cent is easily obtainable. A number of tests are known, which show that mean effective pressures of 350 psi were obtained continuously, and of about 440 psi for a short time.

When supercharging is used to counteract the effect of a suction vacuum due either to high altitude or to high engine speed, both of which produce a low volumetric efficiency, the power increase through supercharging is about 1.3 times greater than the increase of the volumetric efficiency. For example, an increase of the volumetric efficiency from 70 to 90 per cent, or by 28.5 per cent, gives a 37 per cent increase of power.

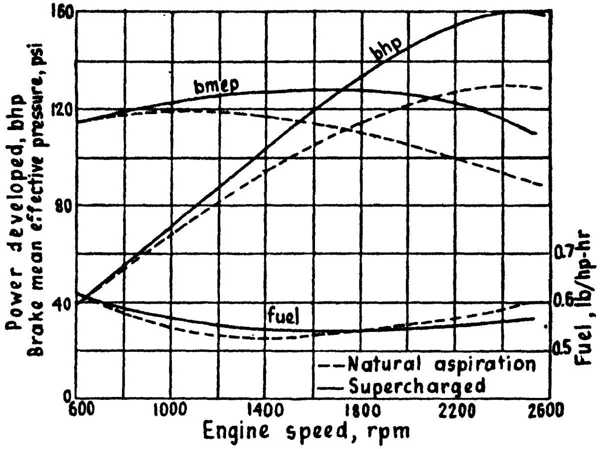

FIG. 19-6.—Effect of supercharging on engine performance.

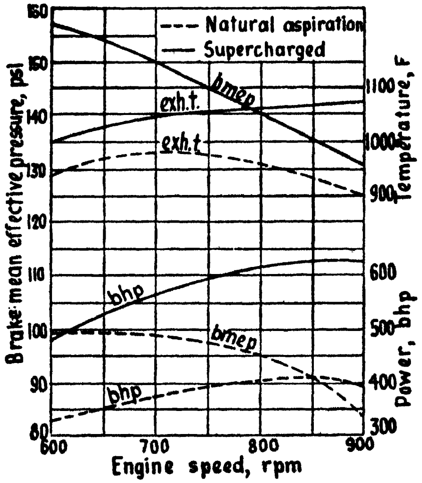

FIG. 19-7.—Effect of supercharging by a Buchi-Elliot turboblower.

Figure 19-6 shows the power gained by supercharging of a 450 cu in. gasoline coach engine at different speeds.[1] Figure 19-7 gives similar data for an eight-cylinder 8-in. × 10½-in.

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[1] BIGGAR, R. E., "Automotive Engines of High Output," a paper presented at an SAE meeting, Jan. 9, 1942, Oklahoma City, Okla.

Cooper-Bessemer compression-ignition oil engine equipped with a Buchi-Elliot turboblower. In this engine the mep when supercharged was limited not by available air but by the fuel-pump delivery.

Finally, Fig. 19-8 presents data about the increase of bmep in two compression-ignition aircraft oil engines as a function of the supercharging pressure.[1] Curve *c* gives the same data in respect to a large oil engine running at 300 rpm with an unchanged valve overlap of 30°.[2]

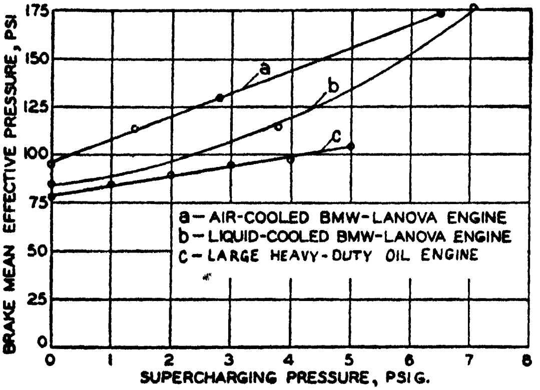

FIG. 19-8.—Effect of supercharging pressure on mean effective pressure.

Mechanical Efficiency.—The increase of friction losses with a supercharger driven by the engine itself is considerably smaller than the power gained through supercharging. As a result the mechanical efficiency, referred to the maximum load, increases with supercharging. Figure 19-9 shows the mechanical efficiencies of a 7-in. × 10-in. six-cylinder compression-ignition Cummins engine at different speeds and mean effective pressures when operating, supercharged with a Roots blower.[3] The mechanical efficiency of the same engine with natural aspiration at 1000 rpm when developing a bmep of 84 psi was found to be 73 per cent. The influence of supercharging upon the mechanical efficiency of two aircraft engines is brought out by Fig. 19-10.

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[1] MALEEV, V. L., *Mech. Eng.*, vol. 63, p. 446, 1941.

[2] *Diesel Power*, vol. 18, p. 877, October, 1940.

[3] KNUDSEN, H. L., Problems and Possibilities of Mechanical Supercharging of Diesel Engines, *Diesel Power*, vol. 19, p. 856, October, 1941.

Fuel Economy.—Owing to better combustion because of increased turbulence, better mixing of the fuel and air, and of an increased mechanical efficiency, the specific fuel consumption in most cases, though not in all, is lowered by supercharging.

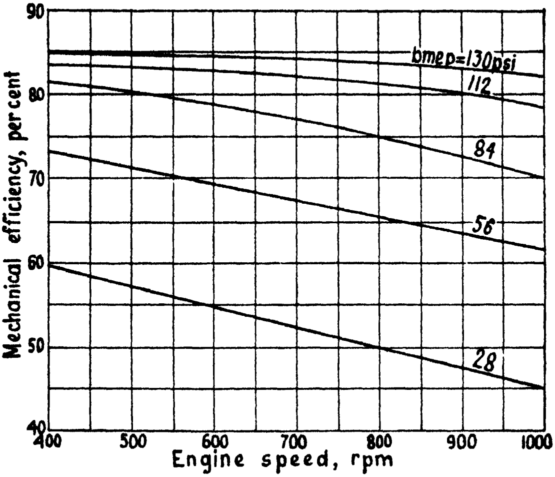

FIG. 19-9.—Mechanical efficiencies of a Cummins supercharged oil engine.

Figure 19-11 gives an interesting comparison of the performance of a three-cylinder 9½-in. × 10½-in. Alco oil engine at 1000 rpm: curve *c*, unsupercharged; curve *d* supercharged with a Roots blower, power increase about 75 per cent, fuel consumption 5 per cent lower; curve *e* supercharged by a Buchi turboblower, power increase about 87 per cent, fuel consumption 13 per cent lower than in the unsupercharged engine.

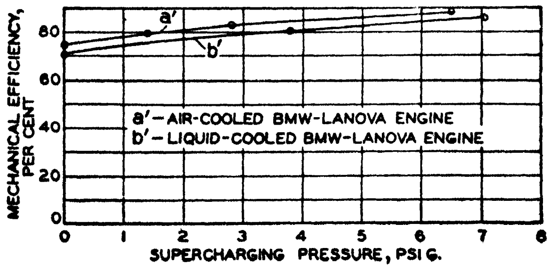

FIG. 19-10.—Effect of supercharging on mechanical efficiency of compression-ignition aircraft engines.

Curves *a* and *b* show also how the performance of this engine was improved by putting in turbulence-creating pistons.

Figure 19-6, on the other hand, shows that the fuel consumption of that particular gasoline engine was improved very little and only at the maximum supercharged horsepower.

Fuel Knock.—In compression-ignition oil engines, increasing the inlet pressure decreases the ignition lag and consequently the rate of pressure rise in the cylinder, which results in an increasing smoothness of operation.[1] On the other hand, in a gasoline engine, if the engine is operated with a compression ratio that causes incipient detonation when supercharging and if a fuel with the same antiknock characteristics is used, in order to prevent objectionable detonation, the compression ratio must be lowered so that the compression pressure of the supercharged engine will remain about the same as before supercharging. This will slightly lower the thermal efficiency, but the power output will be increased, as a greater amount of fuel will be burned. An engine operating with natural aspiration with a compression ratio 7:1, when supercharged should have a compression ratio about 6:1.

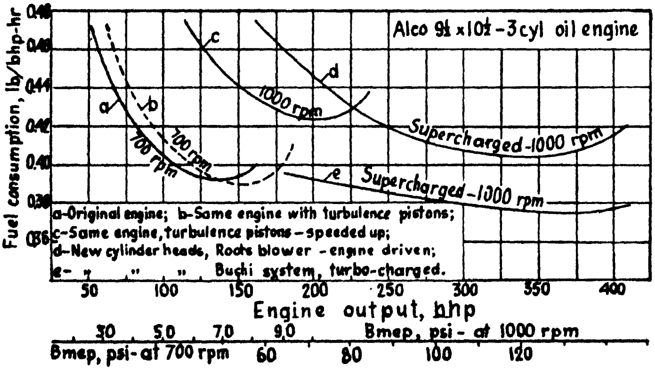

FIG. 19-11.—Performance development of Alco 9½-in. × 10½-in. engine.

19-4. Limitations.—The permissible amount of supercharging depends upon the ability of the engine to withstand the increased pressure and heat stresses.

Pressures.—The increase of the mean effective pressure naturally increases the mean bearing pressures and mechanical friction losses. On the other hand, the maximum pressures and temperatures will go up too. Thus a six-cylinder 12-in. × 15-in. by 650 rpm Enterprise oil engine unsupercharged develops continuously 670 bhp, or a bmep of 80 psi, with a maximum pressure of about 680 psi. When equipped with a Buchi turbine-driven blower, the engine can develop a maximum bmep of 162 psi and a continuous bmep of 125 psi, or a power increase of 56 per cent; the maximum pressure goes up to about 850 psi, the specific fuel consumption goes down from 0.40 lb/hp-hr to 0.375 lb/hp-hr, or an improvement of 6 per cent.

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[1] *NACA Tech. Notes* 509, 1936, p. 8.

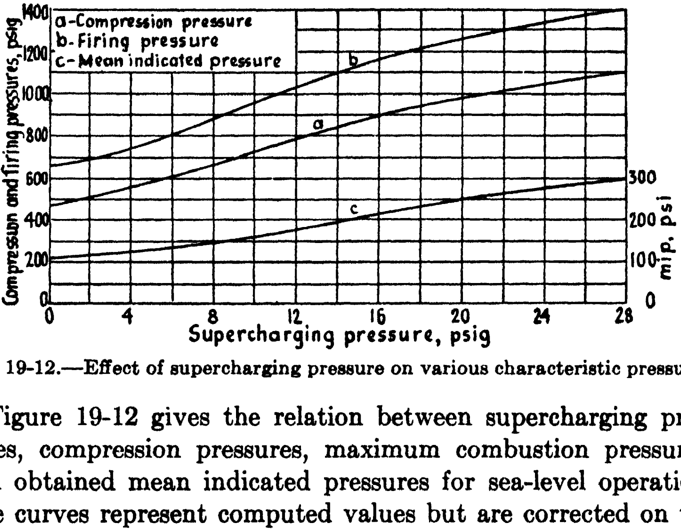

FIG. 19-12.—Effect of supercharging pressure on various characteristic pressures.

Figure 19-12 gives the relation between supercharging pressures, compression pressures, maximum combustion pressures, and obtained mean indicated pressures for sea-level operation. The curves represent computed values but are corrected on the basis of corresponding test data and give an idea of the limit of supercharging.

Temperatures.—The above-mentioned Enterprise engine has an exhaust temperature of 720 F unsupercharged and 960 F supercharged. The Cummins engine mentioned before has exhaust temperatures as shown on Fig. 19-13; Fig. 19-14 shows the temperatures for the Alco three-cylinder 9½-in. × 10½-in. engine at different loads.[1] Finally Fig. 19-15 gives a very complete picture of the performance of a six-cylinder 12½-in. × 13-in. Alco engine, which supercharged is rated 1000 bhp at 740 rpm. The exhaust temperature goes up from 720 F, at 80 psi mep, to 1100 F, at 120 psi mep.

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[1] *Diesel Power*, vol. 19, p. 864, October, 1941.

Remedies.—The high exhaust temperatures have a tendency to burn the piston top and the seat and edges of the exhaust valves. In supercharged engines they are made of materials that better withstand high temperatures. On the other hand, it should be noted that the temperatures given above are obtained in engines either specially built for supercharging or properly adapted for supercharging; in other engines the temperatures will be still higher.

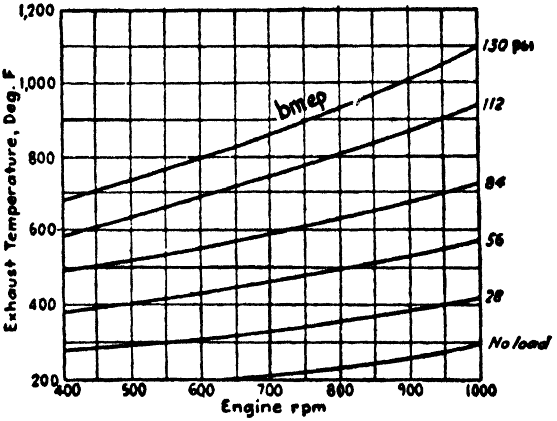

FIG. 19-13.—Exhaust temperatures of a Cummins supercharged oil engine.

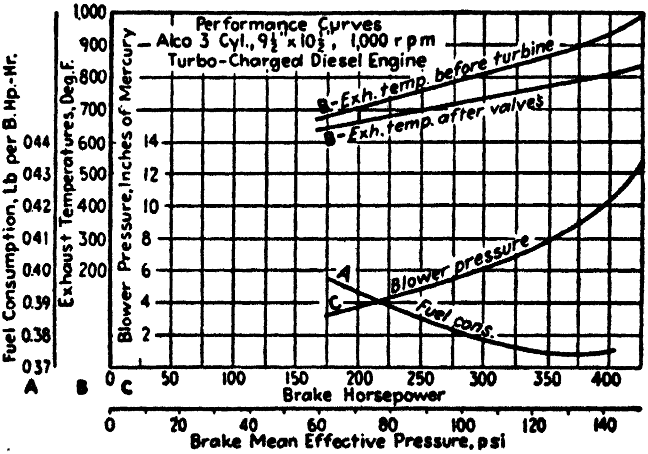

FIG. 19-14.—Performance curves of an Alco oil engine.

Overlap.—One of the main differences between supercharged engines and engines with natural aspiration is the greater overlap of the exhaust and intake-valve timing required by the supercharged engines. Tests with spark-ignition engines have shown that an increase of the overlap from 28 to 90° crank travel materially contributes to scavenging of the combustion space, cooling the exhaust valves, seats, and piston tops and at the same time increasing the horsepower output 10 to 12 per cent.[1] In com-

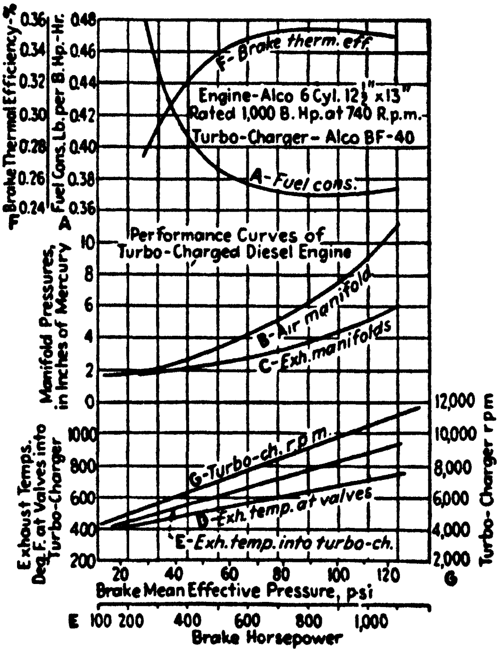

FIG. 19-15.—Performance of a 1000-hp Alco oil engine.

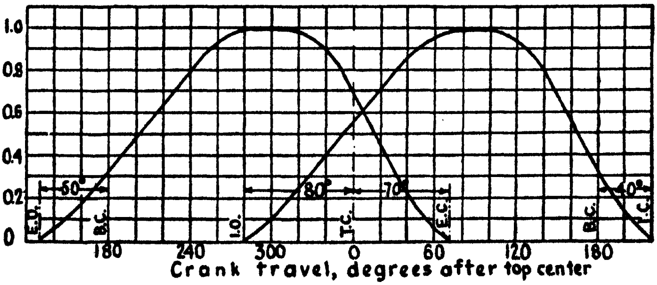

FIG. 19-16.—Cam lifts and valve overlap of a supercharged oil engine.

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[1] *Z. Ver. deut. Ing.*, vol. 85, No. 20, p. 468, 1941.

pression-ignition oil engines a greater angle of overlap is used, which requires excess air to be delivered from the supercharger to take care of the air escaping through the exhaust valve. This excess may vary from about 40 to 80 per cent of the piston displacement, depending upon the amount of overlap, the shape of the cams, supercharger pressure, and engine speed. The overlap is made from 80°, up to 160°. Figure 19-16 shows valve lifts of a 10-in. × 12-in. oil engine operating at 750 rpm; the overlap is $80 + 70° = 150°$.[1] The beneficial influence of a large overlap is a thorough scavenging of the combustion space. The theoretical increase of power due to thorough scavenging, such as is obtained through an overlap of the valve timing, was brought out by the upper curve of Fig. 5-5.

Cooling by Oil.—Another method of cooling the piston top of a supercharged engine is by directing a stream of lubricating oil on it from underneath. The oil is recooled by circulating it through an oil cooler.

Booster Pressures.—The higher the supercharger, or booster, pressure is raised, the more air will be forced into the cylinder and the greater will be the power-output increase. At the same time a greater supercharger pressure absorbs more power to produce it. The polytropic compression of the air has an exponent of about 1.6 to 1.8 for centrifugal superchargers and up to 2.0 for Roots blowers. The increase of temperature during this compression is considerable, from 150 to 200 F. In order to raise the charge efficiency, the air from the supercharger blower is sometimes cooled in a special intercooler before it is admitted to the engine. The power to drive the booster pump increases rapidly with an increase of the compression ratio p_2/p_1; and with a high booster pressure p_2 the power to drive the supercharger may absorb all the gain received by the engine.

No definite data can be given as to the best value of p_2, which is usually found by running a number of comparative tests. In general, booster pressures of the order of 4 to 7 psig, seldom higher, are used. Some idea of the booster pressures used may be obtained from Figs. 19-8, 19-10, and 19-15, which show the influence of the booster pressure on the power output within the usual pressure limits. Figure 19-12 shows the calculated increase of indicated pressure without deduction of the power to drive the booster pump. With this deduction the curve of mean effective pressure would run below that of mip and have a maximum around 10 to 14 psig.

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[1] *Diesel Power*, vol. 19, p. 81, January, 1941.

19-5. Power Requirements.—The work required to compress and deliver a given amount of air in a reciprocating compressor may be computed using as a basis equation (2-68).

In a centrifugal compressor, the air expands due to friction. Figure 2-7 does not represent the correct *p-v* relation, and the expression for work W_k, lb-ft, becomes

$$W_k = 144Wp_1v_1\left(\frac{k}{k-1}\right)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \tag{19-1}$$

where W is the weight of air, lb,
k is the ratio of specific heats, c_p/c_v,
p_1 is the initial pressure, psi,
p_2 is the discharge pressure, psi,
v_1 is the initial specific volume, cu ft/lb,
n is the compression exponent.

The value of n depends upon the amount of heat generated by fluid and mechanical friction within the supercharger. Most of this heat is present in the air at the supercharger outlet. This heat is in addition to the heat added to the air by the compression process. For centrifugal superchargers n varies from 1.6 to about 1.7.

Substituting in equation (19-1) the value RT_1 for p_1v_1, taking W as pounds per minute, using $R = 53.35$ and $k = 1.4$, and dividing by 33,000 gives the indicated horsepower required

$$N_{ic} = 0.00565WT_1[(p_2/p_1)^{\frac{n-1}{n}} - 1] \tag{19-2}$$

For a Roots-type supercharger the power required may be also expressed as

$$N_{ic} = 144V(p_2 - p_1)/33{,}000 = 0.00436V(p_2 - p_1) \tag{19-3}$$

where V is the supercharger displacement, cfm, and may be found as the product of the area enclosed between the rotor and housing wall, Fig. 19-1*a*, the length of rotors, the number of lobes, and rotor speed, rpm.

Another method is to compute the work of supercharging by the equation

$$W_k = 778W(h_2 - h_1) \tag{19-4}$$

where h_2 is the enthalpy of air leaving the supercharger, Btu/lb,
h_1 is the enthalpy of air before the supercharger, Btu/lb.

Enthalpies h_1 and h_2 may be determined using Table 2-2 in connection with equation (2-30). The final temperature t_2 may be either given from direct measurements or computed using equation (2-52) for a given or assumed value of n.

Taking W as pounds per minute and dividing by 33,000, equation (19-4) gives the horsepower required

$$N_{ic} = 0.0236W(h_1 - h_2) \tag{19-5}$$

19-6. Problems.—1. An aircraft engine develops 180 bhp at sea level and has a suction vacuum of 0.8 psi and exhaust back pressure of 0.65 psi. Determine the approximate volumetric efficiency and the supercharger pressures necessary to obtain 90 per cent of sea-level rating at a 15,000-ft altitude and 80 per cent at a 30,000-ft. altitude.

2. Work problem 1 for 88 per cent of sea-level rating at an altitude of 20,000 ft and for 75 per cent at a 35,000-ft altitude.

3. The maximum compression pressure permissible for a certain fuel is 200 psi; for a given engine it is obtained with a compression ratio of 7.25. (*a*) Determine: the relative changes in the suction pressure, volumetric, charge, and ideal efficiencies, and power developed with the same compression pressure with compression ratios 6.5, 5.5 and 4.5; (*b*) plot the results.

4. The maximum compression pressure permissible for a certain fuel is 145 psia and for a certain engine is obtained with a compression ratio of 6.0. Determine the relative changes in the suction pressure, the volumetric, charge, and ideal efficiencies, and power developed with the same compression pressure with compression ratios 4.5 and 7.5.

5. A centrifugal supercharger compresses air of standard conditions. Assuming the exponent of the compression to be 1.65, find: (*a*) the work of delivering and (*b*) the final temperature for compressing 100 lb pm air to 6, 9, and 12 in. Hg; plot the results.

6. Work problem 5 for 7, 10.5, and 14 in. Hg.

7. A centrifugal supercharger must deliver air of atmospheric pressure when taking it in at an altitude. Assuming the exponent of the compression to be 1.65, find: (*a*) the work of delivering 1 lb air and (*b*) the final temperatures for altitudes of 15,000, 25,000, and 35,000 ft and the corresponding temperatures; plot the results.

8. Work problem 7 assuming an exponent $n = 1.75$ and for altitudes of 18,000, 27,000, and 36,000 ft.

9. A Roots blower compresses 1 lb air from standard conditions; the exponent of the compression is 1.7. Find: (*a*) the work to be done and (*b*) the final temperature for delivering the air against pressures of 3, 5, and 7 psig. Compare with adiabatic compression, and plot the results.

10. Work problem 9 assuming an exponent $n = 1.85$ and supercharger pressures of 3.25, 5.25, and 7.25 psig.

11. A Roots-type supercharger compresses air of standard conditions; the exponent of the compression is 1.9. Find: (*a*) the horsepower input for 60 lb pm for supercharger pressures of 4 and 6 psig; (*b*) the amount of heat to be extracted by an air cooler if the air is to be cooled to 80 F.

12. Work problem 11 using an exponent $n = 1.95$ and a delivery of 80 lb pm.

13. For the supercharger of problem 5 find: (*a*) the diameters of the intake and discharge pipes if the allowable air velocities are 350 fps in the intake and 500 fps in the discharge; and (*b*) the discharge velocities for the different supercharger pressures.

14. For the supercharger of problem 11 find: (*a*) the diameter of the intake and discharge lines if the allowable air velocities are 400 fps in the intake and 600 fps in the discharge lines; and (*b*) the discharge velocity for the higher pressure.

CHAPTER 20

TWO-STROKE EXHAUST AND SCAVENGING

20-1. General Remarks. *Porting.*—The determination of the proper size, shape, and arrangement, in short, of the exhaust and scavenge ports, is called *porting*. The performance of a two-stroke engine, meaning its maximum economical mean effective pressure and specific fuel consumption, depends upon the type of scavenging used as discussed in Chapter 15, Table 15-1, but also to a great extent upon its porting. The higher values in Table 15-1 are for engines with a good porting; the lower ones, for those with a poorer porting.

The errors of design most frequently encountered are:[1]

1. Short circuiting. A large amount of scavenge air passes directly to the exhaust ports without displacing residual gases. The shape or direction of the ports must be changed.
2. Insufficient exhaust lead. There is not enough time for the exhaust gases to escape and for the pressure in the cylinder to drop sufficiently before the scavenge ports are opened; residual gases enter the scavenge ports and mix with the air. The crank travel angle between the opening of the exhaust and scavenge must be increased or the scavenge pressure raised.
3. Too late closing of the scavenge ports. If these ports are left open too long, the pressure in the cylinder due to the piston movement will exceed the scavenge pressure, and part of the air will be pushed back into the air receiver.
4. Excessive exhaust-port or valve area or a slow closing of the valve prevents the building up of the pressure in the cylinder and does not allow a sufficient supercharging.

Timing.—Table 20-1 gives the range of port timing in degrees of crank travel before or after dead center, respectively, as found from comparing these data for a large number of engines in operation. The lead and supercharge angles increase with the engine speed and decrease with the supercharge pressure.

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[1] Schweitzer, P. H., Porting of Two-stroke-cycle Diesel Engines, *Diesel Power*, vol. 20, No. 5, May, 1942, p. 401.

Variable Flow.—If the area A of a port changes with the time t, but the velocity v can be considered constant, then the volume V discharged is

$$V = \mu v \int_{t_m}^{t_n} A\,dt \tag{20-1}$$

TABLE 20-1.—RANGE OF PORT TIMING
(Degrees of crank travel)

Type of engine	Exhaust		Lead angle, deg	Scavenge		Supercharging, deg
	Opens	Closes		Opens	Closes	
Crankcase scavenging	60–76	60–76	10–20	45–60	45–60	0
Return-flow scavenging, engine piston	60–76	60–76	10–20	45–60	45–60	0
Return-flow scavenging, separate pump, check valve	59–75	59–75	10–25	33–60	65–80	10–20
Uniflow, port scavenging, one piston	80–90	38–59	20–40	38–55	38–55	0–10
Uniflow, opposed pistons	50–67	50–67	10–20	50–65	50–65	10–20

The quantity $\int_{t_m}^{t_n} A\,dt$ is called the *time integral* of the area and is found by plotting A as ordinates against t as abscissas, Fig. 20-1,

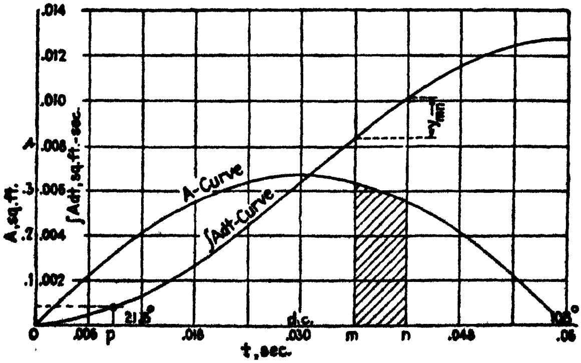

FIG. 20-1.—Opening-area and time-integral diagram.

which represents a case discussed in Example 20-1. As a unit for t it is convenient to take the time element corresponding to about 10° of crank travel. The time integral of the port area for the whole period, from $t = 0$ to the time $t = t_c$ is the area under the A curve. For a period between the time t_m and t_n the time integral $\int_{t_m}^{t_n} A\,dt$ is the area enclosed by the curve A, the axis of t, and the two ordinates drawn through m and n. This area is crosshatched in Fig. 20-1.

It is also convenient to plot on the same base a curve whose ordinates respresent $\int A\,dt$ from $t = 0$. By means of this curve the value of $\int_{t_m}^{t_n} A\,dt$ is found as the difference of ordinates for the points m and n, segment y_{mn}, and the volume discharged would be vy_{mn}.

The gas velocity v may be determined from one of the expressions given above, (18-2) or (18-4).

20-2. Exhaust Calculations.—As mentioned, the scavenge-air receiver should not be put into communication with the engine cylinder until the pressure in the latter has fallen to a pressure equal to that of the scavenge air.

During the exhaust release, both the port area and the velocity change; the port opening increases, Fig. 20-1, and the gas velocity decreases because the pressure difference gradually drops. The amount of gas discharged must be computed from the expression

$$V = \int_0^{t_1} \mu A v\,dt \tag{20-2}$$

where both A and v are functions of the time t. The computation can be greatly simplified if it is noted that from $t = 0$ to $t = t_1$ the port area can be considered proportional to time

$$A = kt \qquad \text{and} \qquad A_1 = kt_1 \tag{20-3}$$

Assuming, as a first approximation, that the velocity drop from v_0 to v_1 is also proportional to time, which means

$$v = v_0 - (v_0 - v_1)t/t_1 \tag{20-4}$$

Substituting expressions (20-3) and (20-4) in expression (20-2) and completing the integration give, in cubic feet,

$$V = \mu A_1 t_1 (v_1 + v_0/2)/3 \tag{20-5}$$

The necessary exhaust calculations can be made using expression (20-5) combined with one of the expressions given for the gas velocity and assuming that the change of state of the exhaust gases can be considered adiabatic.

Discharge Coefficient.—The value of the velocity coefficient φ for the flow of gases through ports is approximately the same as for the flow through valves, about 0.85 to 0.95. The value of the contraction coefficient α lies between 0.65 and 0.85, being smaller when the edges are sharp, *e.g.*, when the ports are drilled, and increasing when the number of ports is small and when they are cast with rounded approach edges, as in Fig. 15-1*c*.

Thus the discharge coefficient $\mu = \alpha\varphi$ may vary from about 0.55 to 0.80, with an average value of about 0.67.

Example 20-1.—Engine data: 10-in. bore, 15-in. stroke, 300 rpm, compression ratio 15; exhaust ports are uncovered by the piston 17 per cent before the end of the stroke; connecting rod to crank ratio is 5:1; charge temperature is 144 F. The cylinder pressure is 60 psia at the point at which the ports begin to uncover; the back pressure in the exhaust line is 15 psia. Find the crank position when the cylinder pressure drops to the scavenge pressure of 17 psia.

Piston displacement is $0.7854 \times 10^2 \times 15/1728 = 0.682$ cu ft
Compression space is $0.682/(15 - 1) = 0.049$
Total volume of gases $= 0.731$ cu ft

At the point of exhaust release the volume of the gases is

$$V_0 = 0.049 + 0.682 \times (1 - 0.17) = 0.615 \text{ cu ft}$$

The same volume will be at the beginning of the compression.

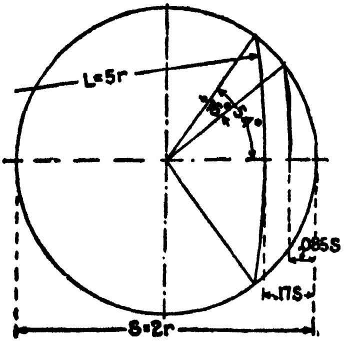

FIG. 20-2.—Timing diagram of a two-stroke engine.

Figure 20-2, drawn with the given data, shows that the ports are uncovered 54° before dead center and closed 54° after dead center.

The temperature at the point of uncovering the exhaust ports is, from the charge temperature, taking into consideration the difference of pressures

$$T_0 = (144 + 460)60/17 = 2130 \text{ R}$$

With a fair approximation the gases can be assumed to have the thermal properties of air.

The weight of the gases in the cylinder is, from equation (2-2),

$$W_0 = 60 \times 144 \times 0.615/(53.3 \times 2130) = 0.0468 \text{ lb}$$

Assuming first that the lead angle will be 16° until the pressure drops to 17 psi, the volume of the gases at this point is from Fig. 20-2

$$V_1 = 0.049 + 0.682 \times (1 - 0.085) = 0.673 \text{ cu ft}$$

The temperature, from expression (2-52), with $k = 1.35$ for the higher temperature range is

$$T_1 = 2130(17/60)^{0.259} = 1545 \text{ R}$$

and the weight of gases left in the cylinder is

$$W_1 = 17 \times 144 \times 0.673/(53.3 \times 1545) = 0.0202 \text{ lb}$$

Thus the weight discharged is

$$W_d = W_0 - W_1 = 0.0468 - 0.0202 = 0.0266 \text{ lb}$$

The next thing to find is the rate of discharge per unit of port area. At the initial pressure of 60 psia, the throat pressure

$$p_2 = 60 \times 0.53 = 31.8 \text{ psi}$$

and the throat temperature

$$T_2 = 2130(31.8/60)^{0.259} = 1804 \text{ R}$$

From expression (18-5), with $\varphi = 0.85$, and $13{,}430/60 = 224$, and $c_p = 0.27$ at this temperature level, the throat velocity is

$$v_2 = 224 \times 0.85 \sqrt{(2130 - 1804) \times 0.27} = 1786 \text{ fps}$$

The specific volume at the throat is

$$v_2' = 53.3 \times 1804/(31.8 \times 144) = 21.0 \text{ cu ft/lb}$$

and the rate of discharge is with a coefficient of contraction $\alpha = 0.70$

$$q_2 = 1786 \times 0.70/21.0 = 59.5 \text{ lb/sq ft-sec}$$

At the lower pressure of 17 psia the throat pressure is equal to the back pressure of 15 psia, and the throat temperature

$$T_3 = 2130(15/60)^{0.259} = 1487 \text{ R}$$

and the throat velocity

$$v_3 = 224 \times 0.85 \sqrt{(1545 - 1487) \times 0.27} = 753 \text{ fps}$$

The specific volume is

$$v_3' = 53.3 \times 1487/(15 \times 144) = 36.7 \text{ cu ft/lb}$$

and the rate of discharge is

$$q_3 = 753 \times 0.70/36.7 = 14.34 \text{ lb/sq ft-sec}$$

If the assumption is made that the rate of discharge per unit area changes from the higher value to the lower one uniformly with respect to time, then equation (20-5) can be used with W_d instead of V and q instead of v, giving

$$0.0289 = A_s t_s(14.34 + 59.5/2)/3$$

from which $A_2t_2 = 0.001966$. From Fig. 20-1, approximately,

$$\int A\,dt = \tfrac{1}{2}A_2t_2 = 0.00098 \text{ sq ft-sec}$$

and this value corresponds to point p, about 16° after the exhaust ports begin to open.

20-3. Scavenging.—The main prerequisite for the efficient performance of a two-stroke engine is proper scavenging. This can be obtained only if the scavenge air is admitted in a solid column which pushes the burned gases before it into the exhaust pipe without mixing with them. This condition is equivalent to the requirement of avoiding whirls and eddies in the scavenge air. Experiments have shown that the best results are reached when the scavenge-air pressure is small and constant.

The smallest permissible scavenge-air pressure depends upon the size and resistance of the scavenge ports and the duration of their opening. To admit the necessary amount of air without unduly raising the scavenge pressure the ports must be as large as possible and the duration of opening must be adequate.

Example 20-2.—Find the air velocity and air flow in cfpm for a cylinder pressure $p_0 = 15.4$ psia, scavenge pressure $p = 16.9$ psia, and temperature of scavenge air $t = 100$ F.

Assuming $\varphi = 0.90$, $\alpha = 0.70$, and having $T_1 = 100 + 460 = 560$ R, the temperature T_0 is found, with $k = 1.41$ for this temperature range,

$$T_0 = 560(15.4/16.9)^{0.291} = 545 \text{ R}$$

and

$$w_g = 15.4 \times 144/53.3 \times 545 = 0.0765 \text{ lb/cu ft}$$

and the air velocity, by expression (18-2),

$$v = 3780 \times 0.90 \sqrt{(16.9 - 15.4)/0.0765} = 15{,}050 \text{ fpm}$$

and from expression (18-6) the amount discharged

$$V = 0.70 \times 15{,}050 \times A = 10{,}530A \text{ fpm}$$

This example shows that even with a small scavenge pressure the velocity becomes very high, higher than velocities used in valve ports of four-stroke engines.

Scavenge-air Pressure.—The necessary scavenge pressure p can be determined by a method similar to the exhaust calculations, using equation (20-5) and finding the pressure differences necessary to produce the velocities v_0 and v_1 from expression (18-2) or from Fig. 18-6. However, p thus obtained will not be quite correct, being below the actual value on account of neglecting the effect of discharging the excess air through the exhaust ports. A convenient cut-and-try method of correcting the preliminary value of p can be explained best by an example.

Example 20-3.—The engine discussed in Example 20-1 was changed to uniflow scavenging, Fig. 15-15. Determine the necessary scavenge-air pressure using the following additional data: the engine has two exhaust valves $4\frac{1}{4}$ in. diameter, maximum lift $\frac{7}{8}$ in., opening 54° before, and closing 26° after dead center; the valve-lift curve is harmonic, *i.e.*, the lift plotted against the crank angle is a sine curve; the scavenge ports open 40° before and close 40° after dead center; they occupy 0.42 of the circumference; the temperature of the scavenge air is 100 F; excess air is 45 per cent of piston displacement referred to free air.

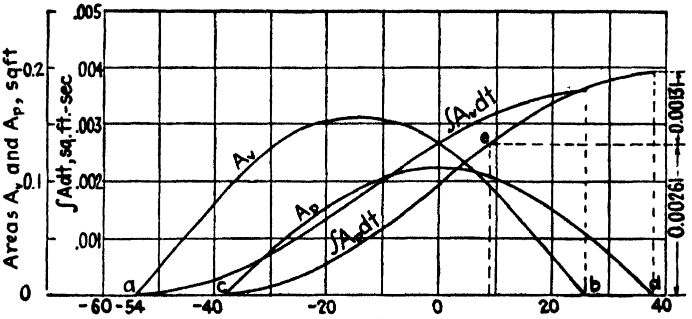

FIG. 20-3.—Area and time-integral curves for Example 20-3.

The volume of the scavenge air at standard pressure and temperature per revolution is

$$V = 0.682 \times 1.45 = 0.989 \text{ cu ft}$$

The maximum scavenge-port area

$$A_p = 0.42 \times \pi \times 10 \times 0.096 \times 55/144 = 0.132 \text{ sq ft}$$

The maximum exhaust-valve area

$$A_v = 2 \times \pi \times 4.25 \times 0.875/144 = 0.162 \text{ sq ft}$$

The consecutive values of the exhaust-port and scavenge-valve areas were plotted against the crank angles on Fig. 20-3.

The number of seconds corresponding to 1° crank travel

$$t = 60/(300 \times 360) = 0.000556 \text{ sec}$$

In Fig. 20-3 values of $\int A\,dt$ for both exhaust and scavenge areas were plotted. Since the lift of the exhaust valves is harmonic, the mean area is one half of the maximum, and the final value of the area-time integral is

$$\int_a^b A_v\,dt = 0.162 \times 0.5 \times 0.000556 \times 80 = 0.0036 \text{ sq ft-sec}$$

Since the port-area curve is nearly a parabola, the mean area is two-thirds of the maximum, and the final value of the area-time integral is

$$\int_c^d A_p\,dt = 0.132 \times 0.667 \times 0.000556 \times 80^\circ = 0.00392 \text{ sq ft-sec}$$

A first approximation of the air pressure in the scavenge receiver required by the conditions of the problem is obtained assuming that the scavenge air is delivered against a constant pressure of 15 psia.

The temperature of the scavenge air expanded adiabatically to 15 psia is

$$T = 560(15/17)^{0.291} = 540 \text{ R}$$

and the volume

$$V = 0.989 \times 14.7 \times 540/(15 \times 520) = 1.01 \text{ cu ft}$$

The average actual velocity, with a contraction coefficient $\alpha = 0.75$, would be

$$v_1 = 1.01/(0.0036 \times 0.75) = 373 \text{ fps}$$

Dividing v_1 by a velocity coefficient of 0.85 gives the theoretical velocity, $v_2 = 439$ fps, and the corresponding pressure difference from Fig. 18-6 is 1.5 psi, giving a scavenge-air pressure $p = 15 + 1.5 = 16.5$ psia. This figure is slightly too low, for the effect of discharging the excess air through the exhaust valves is not taken into account.

Assume for a trial a slightly higher scavenge pressure of $p = 16.6$ psi. The theoretical velocity from Fig. 18-6 is 440 fps, and the actual average velocity

$$v_2' = 440 \times 0.85 = 374 \text{ fps}$$

The value of $\int A_p\,dt$ to fill the cylinder volume $V = 0.731$ cu ft, Example 20-1, is therefore, from formula (20-1),

$$\int_0^t A_p\,dt = 0.731/(374 \times 0.75) = 0.00261 \text{ fps}$$

This corresponds to point e in Fig. 20-3 and shows that with a scavenge pressure of 16.6 psi the cylinder would be filled with scavenge air at about 9° after the dead center. From this point until the point d where the scavenge ports are closed, the following phenomena occur simultaneously:

1. Scavenge air is entering the cylinder, raising its pressure.
2. Part of scavenge air is escaping through the exhaust valves.
3. The piston is rising, reducing the volume of the cylinder content and tending to raise the pressure.

If the pressure p_1, which exists in the cylinder at point d, equals the scavenge pressure, which is its upper limit, the amount of air introduced into the cylinder during the period under consideration would be

$$V_d = \tfrac{2}{3} \int_a^b A\, dt\, v_1'$$

From Fig. 20-3 the time-area integral is 0.00131; thus

$$V_d = \tfrac{2}{3} \times 0.00131 \times 374 = 0.326 \text{ cu ft}$$

Adding V_d to the volume of 0.731 cu ft already introduced gives 1.057 cu ft, sufficiently close to and on the safe side of the required 1.01 cu ft. Thus the assumption of 16.6 psi for the scavenge pressure is satisfactory.

The exact value of p_1 is of great importance, since upon it depends the amount of charge and, consequently, the load capacity of the cylinder. In building a new engine, p_1 is adjusted experimentally by changing the width of the scavenge ports and checking the desired value of p_1 from weak-spring indicator diagrams.

20-4. Accessories.—In general the same accessories are needed for the air intake and exhaust as for four-stroke engines. Engines with separate scavenge pumps as a rule use air filters, and those with reciprocating pumps may need intake silencers if air filters are not used.

Two-stroke engines are very sensitive to the exhaust back pressure and often are equipped with a separate exhaust pipe from each cylinder to a common pit or muffler. Such a down pipe with a water jacket partly welded on, elbow *b*, is shown in Fig. 20-4; the jacket *d* is bolted to the elbow and held at the bottom by a packing *e* and gland *f*.

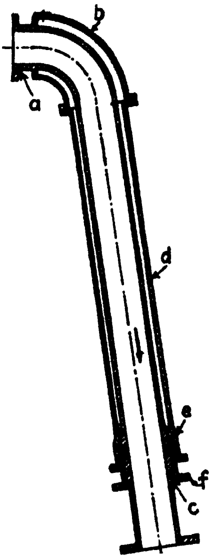

FIG. 20-4.—Two-stroke water-cooled exhaust pipe.

The performance of engines with ports can be materially improved by using guide vanes,[1] as shown in Fig. 20-5 for an opposed-piston engine.

An interesting and comprehensive picture of the pressure events in an experimental M.A.N. engine with backflow scavenging, Fig. 15-8, is presented in Fig. 20-6[2]. The crosshatched contour *e* shows the area of the exhaust ports uncovered by the piston, and that of *s*, the area of the scavenge ports. The

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[1] SORENSEN, E., Strömungsfragen am Dieselmotor, *Z. Ver. deut. Ing.*, vol. 84, p. 884, 1940.

[2] *Z. Ver. deut. Ing.*, vol. 84, p. 137, 1940.

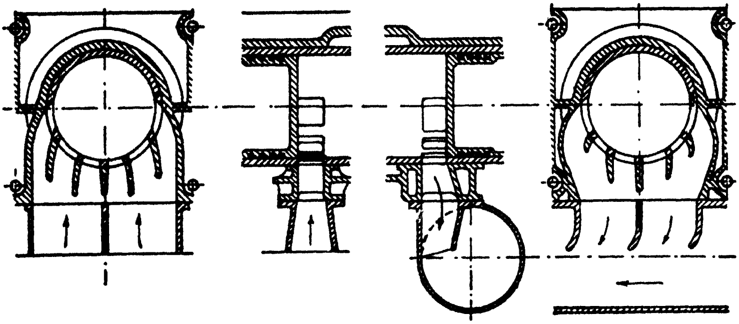

FIG. 20-5.—Scavenge and exhaust ports with guide vanes.

horizontal lines show the average pressures in the exhaust muffler and scavenge receiver, whereas the fluctuating pressures were recorded in the exhaust and scavenge ports outside the engine

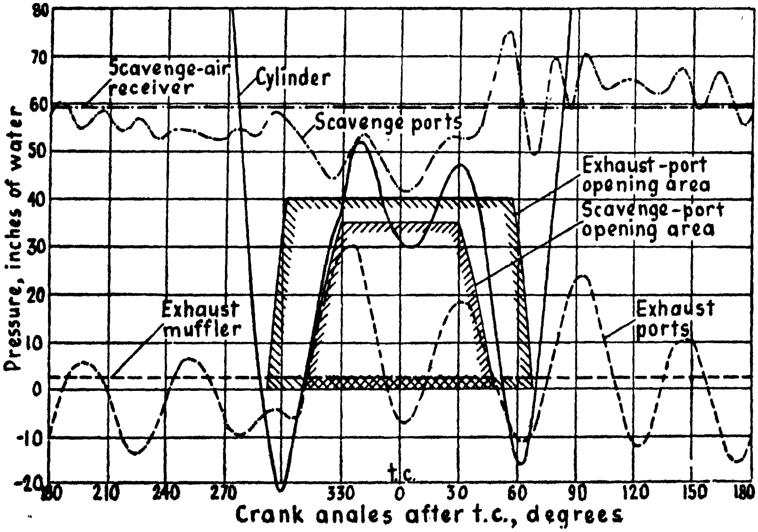

FIG. 20-6.—Pressures in the cylinder, scavenge, and exhaust ports of an experimental engine with M.A.N.-return scavenging, at 240 rpm and 787 fpm piston speed.

cylinder. While the ports are open, the main pressure fluctuations in them follow the pressure fluctuations in the cylinder, which are due to the inertia of the escaping gases. The sinusoidal character of the pressure curve in the exhaust after closing of the ports is quite evident, while in the scavenge ports it is distorted by the discharge from the scavenge pump. The cylinder-pressure waves represent air waves in a motoring diagram without combustion.

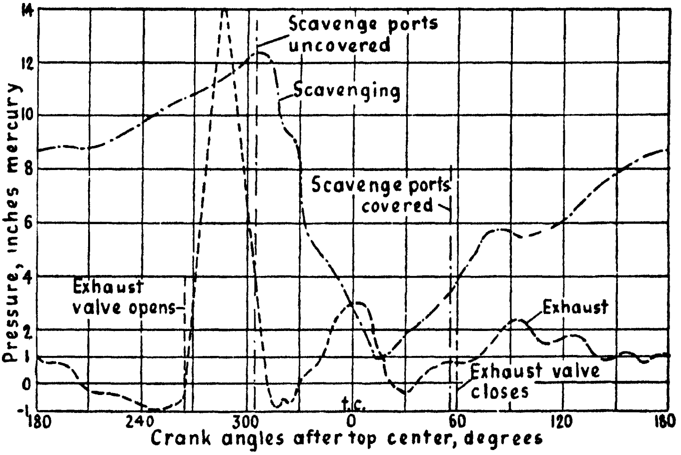

Fig. 20-7.—Pressures in a 15-hp compression-ignition oil engine at 1200 rpm

Figure 20-7 shows the exhaust and scavenge pressures in the corresponding pipes next to the cylinder of a small high-speed compression-ignition engine with uniflow scavenging, Fig. 15-5. Owing to the short connection between the scavenge pump and the cylinder, no receiver being used, there is hardly any inertia influence after the first drop of the pressure.

These brisk exhaust pressure variations explain the necessity of using mufflers and, at the same time, indicate that the length of the exhaust pipe and the volume of gas in it will materially influence the scavenging of the engine.

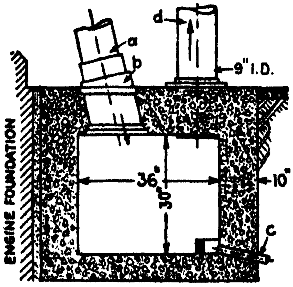

Fig. 20-8.—Exhaust pit for a two-stroke engine.

Mufflers for two-stroke engines do not present any peculiarities, except that it is very essential to have a very small back pressure, as shown in Fig. 20-6.

Figure 20-8 shows the cross section of a concrete exhaust pit for a two-stroke oil engine. Each cylinder has an exhaust pipe *a*, similar to the one shown in Fig. 20-4. After expanding in the pit and losing most of their velocity, the gases ooze out noiselessly through the stack *d*. Any water which may come either from condensation of combustion products of hydrogen or from injection cooling of the exhaust is gradually discharged through a pipe nipple *c*.

A bias-cut end of the exhaust pipe, Fig. 18-32, is also very helpful to reduce the exhaust noise without affecting the back pressure.

20-5. Problems.—**1.** Using Table 20-1 and former data about fuel injection draw a complete timing diagram for a crankcase-scavenging engine running (*a*) at 240 rpm and (*b*) at 400 rpm.

2. Work problem 1 for an engine with a return-flow scavenging that uses the back of the piston for scavenge-air delivery.

3. Using Table 20-1 and former data about fuel injection draw a complete timing diagram for a medium-size return-flow-scavenging engine as shown in Fig. 15-6, running at 240 rpm.

4. Work problem 3 for an engine as shown in Fig. 15-8.

5. Using Table 20-1 and former data about fuel injection draw a complete timing diagram for a uniflow scavenging engine, Fig. 15-15, running at 480 rpm.

6. Using Table 20-1 and former data about fuel injection draw a complete timing diagram for an opposed-piston engine, Fig. 15-11, running at 600 rpm; select all kinematic relations, assuming a bore of 6 in.

7. Using Table 20-1 and former data about fuel injection draw a complete timing diagram for an opposed-piston engine, Fig. 15-13, running at 1200 rpm.

8. Find the necessary lead angle and the scavenge-port length for a return-flow-scavenging engine, using the following data: the 13-in. × 14½-in. engine runs at 320 rpm; the compression ratio is 15.2; the exhaust ports are uncovered 20 per cent before the end of the stroke; the connecting-rod length to crank radius ratio is 4.52:1; the charge temperature is 210 F; the maximum cylinder pressure at the moment of release is 45 psig; the back pressure is 0.8 psig. The scavenge pressure is 2.0 psig; find what percentage of the cylinder circumference the exhaust ports must occupy.

9. If in problem 8 the lead angle is given as 17.5° and the total width of the exhaust ports is 50 per cent of the cylinder circumference, find the minimum scavenge pressure possible to use.

10. (*a*) Find the capacity of the Roots scavenge pump for a 9-in. × 12-in. engine with port scavenging operating at 400 rpm; during scavenging the cylinder pressure is 15.0 psia, the scavenge pressure is 1.75 psig, the temperature of the outside air is 80 F; the scavenge ports are opened 48° before dead center; the connecting rod is 27 in. long, the ports occupy 0.40

of the cylinder circumference; nominal compression ratio $r = 15.4$; also find: (*b*) the scavenge efficiency for these data; and (*c*) the excess air coefficient.

11. (*a*) Find the capacity of the Roots scavenge pump for a 8½-in. × 10-in-6 cylinder engine, rated speed 600 rpm; the cylinder pressure is 2 in. Hg, the scavenge pressure is 6-in. Hg; the outside temperature is 75 F; the temperature drop from the blower to the cylinder is 20 F; the scavenge ports open 52.5° before dead center; the ports occupy 0.35 of the cylinder circumference; (*b*) find the scavenge factor for these data.

12. Find the velocity of scavenge air if the pressure in the scavenge-air receiver is 3.3 psig, the temperature is 105 F, and the pressure in the cylinder is 0.8 psig.

13. Find how much the air velocity will be increased if the pressure difference in problem 12 is doubled by raising the scavenge-receiver pressure.

14. Determine the average area of the scavenge ports and the part of the circumference which they must occupy, with the conditions of problem 12, for a 20½-in. × 27½-in. engine, Fig. 15-6, running at 257 rpm, if a scavenge factor of 1.5 is desired; assume a suitable timing of the scavenge process.

15. Determine the average area of the scavenge ports and the part of the circumference which they must occupy, with the conditions of problem 12, for a 16-in. × 20-in. engine with backflow scavenging, Fig. 15-1*c*, running at 300 rpm.

16. An 8½-in. × 10-in. engine running at 720 rpm has four exhaust valves, 3½ in. diameter, maximum lift $\frac{13}{16}$ in., opening 75° before and closing 45° after dead center; the valve-lift curve is harmonic, see Example 20-3; the cylinder pressure at the moment of release is 92 psia; the scavenge pressure is 4 psig; determine the lead and supercharge angles, also the excess air coefficient. Temperature of charge is 130 F; compression ratio is 15; connecting rod is 22½ in. long.

17. If in problem 16 the lead angle is given as 30°, determine the necessary scavenge pressure to obtain an excess air coefficient of 0.5, leaving all other data unchanged.

18. An opposed-piston 4¾-in. × 8¾-in. × 2 engine, Fig. 15-11, must operate at 1700 rpm; the ports open at 11.4 per cent of the piston stroke before the corresponding dead center; the angle between the cranks is 12°; the connecting rods are 17½ in. long; the cylinder pressure at the point of release is 55 psig. Determine: (*a*) the minimum scavenge pressure to be used; (*b*) the relative amount of exhaust gases escaping during the angle of release; (*c*) the expected scavenge efficiency, and (*d*) the necessary capacity of the scavenge pump for a six-cylinder engine.

CHAPTER 21

COOLING

21-1. General. *Necessity of Cooling.*—Part of the heat developed during combustion flows from the gases to the cylinder walls, raising their temperature. If the wall temperature is allowed to rise above a certain limit, about 300 F, the oil that lubricates the piston begins to evaporate rapidly, and both piston and cylinder may be injured. At the same time high local temperatures in certain parts of the engine, *e.g.*, cylinder head and piston, may cause excessive stresses and cracking of these parts. In gas and gasoline engines hot spots in the combustion space may cause preignition.

For all these reasons, since heat cannot be prevented from flowing to the cylinder walls, it must be carried away. The quantity of heat generated in an engine cylinder varies from about 7000 to 12,000 Btu/hp-hr. Tests show that from 25 to 35 per cent of this heat in stationary and automotive and about 12 to 15 per cent in air-cooled aircraft engines finds its way into the cylinder walls and must be carried away. This heat is often called *jacket loss.*

Methods.—All heat carried away from an engine in the final count is conveyed to the atmosphere. However, the methods of cooling may be divided into two main groups: direct, or air cooling, and indirect, or liquid cooling. These two methods differ in construction details and also in operating conditions—temperature of the cylinder walls. Therefore they will be discussed separately.

21-2. Heat Transfer.—Of the three means of heat transfer, conduction, convection, and radiation, in cooling engine cylinders, conduction plays an important part in carrying the heat through the thin layers of stagnant gas and water in contact with the walls. The rest of the heat exchange is done partly by radiation but chiefly by convection.

The heat flow from one fluid to another, separated by a metal wall, can be best explained using the diagram in Fig. 21-1. The temperature t_a of the gas at point *a* in the interior of the cylinder gradually falls to the value t_b at the surface of the inert gas film. The resistance of this film is very great, a great temperature head, $t_b - t_c$, being required to pass through it by conduction. The temperature head required to cause the flow through the metal wall is $t_c - t_d$. The temperature head required to pass through the outside film is $t_d - t_e$. Its value is comparatively small if the cooling fluid is water and large if it is air. Finally, the temperature drops to t_f of the cooling liquid at some distance away from the wall.

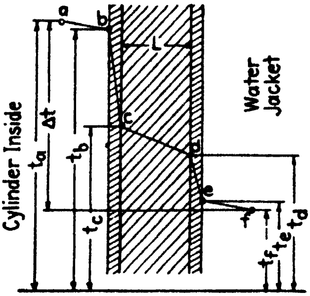

FIG. 21-1.—Heat flow from hot gases to water.

Using the over-all heat-transfer coefficient *U*, instead of h_m, the general equation (12-7) per hour becomes

$$Q = UA(t_1 - t_2) \qquad (21\text{-}1)$$

and *U* can be presented as

$$U = 1/(1/h_1 + L/k + 1/h_2) \qquad (21\text{-}2)$$

where h_1 is the inside surface coefficient, Btu/sq ft-°F-hr,
h_2 is the outside surface coefficient, Btu/sq ft-°F-hr,
L is the thickness of cylinder wall, in.,
k is the thermal conductivity of the metal, Btu/sq ft-°F-hr per inch thickness.

For a cylinder the area *A* in equation (21-1) is the area corresponding to the mean diameter, $A = 0.5(A_o + A_i)$, where A_o is the area corresponding to the outside diameter and A_i to the inside diameter. The difference in the areas absorbing heat from the hot gases and releasing it to the cooling medium can be taken into account by introducing the weighted values for h_1 and h_2, which changes expression (21-2) to

$$U = \frac{1}{A/(A_ih_1) + L/k + A/(A_0h_2)} \qquad (21\text{-}3)$$

The coefficient h_1 varies considerably during the cycle, because of the change of gas pressure and turbulence. Its value can be computed from the previously given expression (12-10). The mean value of h_1 varies from about 50 to 100 and can be estimated if the rates of heat loss and temperature are known.

Figure 21-2 gives the values of the gas film h_1 for the whole cycle of an airless-injection compression-ignition oil engine operating at 1600 rpm for full-load conditions.[1] The pressure and temperature curves complete the picture. The mean value of $h_1 = 62$ Btu/sq ft-°F-hr.

The value of h_2 depends in the first place on whether the outside surface of the cylinder is cooled directly by air or by a liquid.

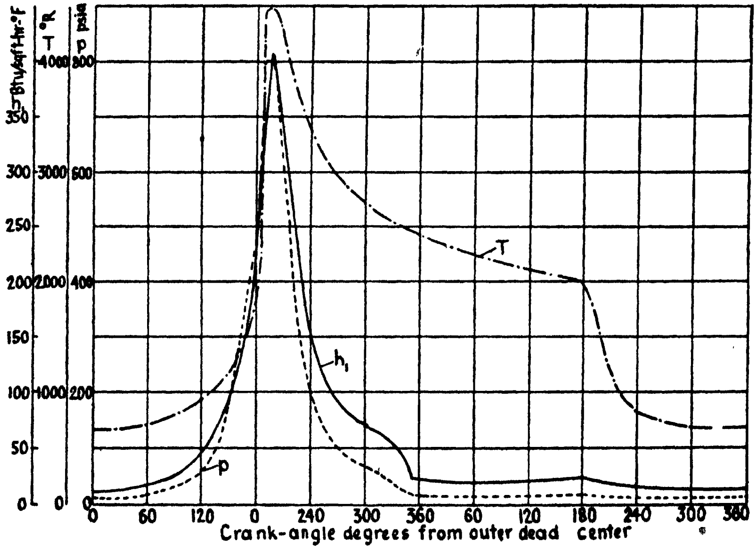

FIG. 21-2.—Temperatures and gas-film coefficients in a compression-ignition oil engine at 1600 rpm.

For the heat exchange between a metal wall and air, h_2 can be computed from the equation

$$h_2 = mv^{0.89} \tag{21-4}$$

where v is velocity of air in feet per minute and the constant $m = 0.01$ for normal conditions and may reach 0.025 if the heat exchange is very efficient, as in automobile radiators.

For the heat exchange between a metal wall and water, h_2 can be determined from the equation[2]

$$h_2 = 60 + b\sqrt{v} \tag{21-5}$$

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[1] VOGT, C. J., Internal Combustion Engine Heat Flow, *Calif. Oil World*, July, 1939.

[2] DUBBEL, H., Oel und Gasmaschinen, p. 66, J. Springer, Berlin, 1926.

where v is the water velocity in feet per minute and b is a constant. For a quiet flow $b = 30$; sharp turns and sudden changes of velocity in the water jackets, by creating turbulences, may increase the value of b up to 60 and even 90.

The thermal conductivity k of metals can be taken from Table 21-1. As can be seen from expression (21-2), the coefficient of conductivity k affects the over-all coefficient U only very little.

TABLE 21-1.—PROPERTIES OF METALS

Alloy	Specific heat, Btu/lb-°F	Average conductivity, $\frac{\text{Btu-in.}}{\text{hr-sq ft-°F}}$ k	Coefficient of linear expansion per °F $a \times 10^6$	Modulus of elasticity in tension, $E \times 10^{-6}$ psi	Poisson's ratio, in./in.
Aluminum (Y alloy)	0.207	1200	14.1	9–10.5	0.33
Brass, yellow	0.090	740	10.4	13–16	0.34
Bronze	0.102	710	9.8	15–17	0.34
Cast iron, gray	0.130	320	6.5	12–14	0.27
Monel metal	0.127	180	8.3	25–26	0.27
Steel, mild	0.110	310	6.5	29–31	0.30

Example 21-1.—Compute the value of h_1 during the expansion and exhaust strokes for a 10-in. × 18-in. gas engine with a mean inside wall temperature of 500 F. For gas temperatures see Fig. 12-2; the compression ratio $r = 5.1$. During a test the engine developed 50 ihp at 250 rpm and showed a thermal efficiency of 28 per cent; heat loss to the walls was 29 per cent of the total heat.

The average surface in contact with the gases, noticing that the liner area changes from a full value to zero and assuming that the combustion space is a cylinder with the height $18/(5.1 - 1) = 4.39$ in., is

$$\begin{aligned} A_i &= (\pi \times 10 \times 18 \times 0.5 + \pi \times 10 \times 4.39 + 0.7854 \times 10^2 \times 2)/144 \\ &= 4.01 \text{ sq ft} \end{aligned}$$

The heat generated, $Q_g = 50 \times 2545/0.28 = 454{,}500$ Btu/hr. The heat lost to the walls $q = 454{,}500 \times 0.29 = 131{,}800$ Btu/hr, and at the same time

$$q = h_1(t_1 - t_2)A\tau \qquad (21\text{-}6)$$

The mean gas temperature for the expansion and exhaust strokes, from Fig. 12-2, is 1400 F, and the time element, two out of four strokes, $\tau = 0.5$. Solving equation (21-6) for h_1 gives

$$h_1 = 131{,}800/[(1400 - 500) \times 4.01 \times 0.5] = 73.0 \text{ Btu/sq ft-°F-hr}$$

It is interesting to check the value of h_1 by formula (12-10). The average piston velocity is

$$c = 18 \times 2 \times 250/12 = 750 \text{ fpm}$$

The average gas pressure may be assumed as 100 psi. Using the average value of $b = 0.0003$ gives

$$h_1 = 0.0003(160 + 750) \sqrt[3]{100^2 \times 1860} = 72.4 \text{ Btu/sq ft-°F-hr,}$$

a very close check.

Example 21-2.—Check the amount of heat carried away by the water jacket in Example 21-1, if the mean temperature of the water is 120 F, the cylinder wall is ⅞ in. thick, and the average water velocity in the jacket is 7 fpm.

From expression (21-5), using an average value of $b = 45$, the outside surface coefficient

$$h_2 = 60 + 45\sqrt{7} = 179$$

For U, equation (21-3) should be used. The outside surface, with

$$D_0 = 10 + 2 \times 0.875 = 11.75 \text{ in.}$$

and with the piston top in contact not with the water, but with the exhaust ports and ribs increasing the cylinder head surface, about 50 per cent,

$$\begin{aligned} A_o &= (\pi \times 11.75 \times 18 \times 0.5 + \pi \times 11.75 \times 4.39 + 0.7854 \times 11.75^2 \times 1.5)/144 \\ &= 4.57 \text{ sq ft} \end{aligned}$$

and

$$A = (4.01 + 4.57)/2 = 4.29 \text{ sq ft}$$

Using $h_1 = 73$ from Example 20-1 and taking from Table 21-1 the heat conductivity for cast iron $k = 320$, gives

$$U = \frac{1}{4.29/(4.01 \times 73) + 0.875/320 + 4.29/(4.57 \times 179)} = 44.2 \text{ Btu/sq ft-°F-hr}$$

From expression (12-7) with $\tau = 0.5$, since during the intake and compression strokes the temperature difference and hence the heat exchange are very small,

$$Q = 44.2 \times 4.29(1400 - 120) \times 0.5 = 121{,}400 \text{ Btu/hr,}$$

a sufficiently close check with 131,800 Btu/hr, found by test and used in Example 21-1.

Cylinder Temperatures.—Calculations similar to the last example show that the accuracy of heat-transfer coefficient U from expression (21-2) is not very great, about ±5 per cent. However, this method of splitting the over-all coefficient in three components is valuable for determining the average temperatures t_2 of the inside surface of the cylinder and t_3 of its outside surface.

The gas, which has a temperature t_1, loses to the inside surface of the wall the heat amount

$$Q = h_1 A_i (t_1 - t_2)\tau \tag{21-6}$$

which gives

$$t_2 = t_1 - Q/h_1 A_i \tau \tag{21-7}$$

The same quantity of heat travels through the wall with a temperature drop of $t_2 - t_3$; therefore

$$Q = (k/L) A (t_2 - t_3)\tau \tag{21-8}$$

This gives for the temperature drop

$$t_2 - t_3 = QL/kA\tau \tag{21-9}$$

where A is the mean between the inside surface A_i and outside one A_o. Thus the inside temperature t_2 found by expression (21-7) will permit t_3 to be determined.

Another approach to finding the cylinder-wall temperature is first to find t_3 from the equation

$$Q = h_2 A_o (t_3 - t_4)\tau \tag{21-10}$$

which gives

$$t_3 = t_4 + Q/h_2 A_o \tau \tag{21-11}$$

and then to determine t_2 by means of equation (21-9).

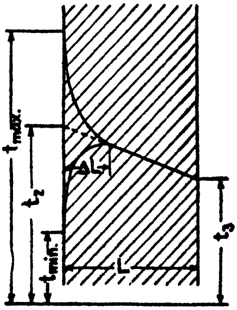

FIG. 21-3.—Temperature fluctuation of the inner cylinder surface.

The temperature t_2 of the inside surface of the cylinder wall is not constant during a cycle but fluctuates following the variation of the gas temperature. Figure 21-3 gives a diagram of the temperature distribution through the wall. The temperature of the inner surface goes up to t_{max} during combustion, and drops to t_{min} toward the end of the suction stroke. Eichelberg[1] found for a two-stroke oil engine that at full load the fluctuation above the average value, $t_{max} - t_2$, is about 25 F and below it, $t_2 - t_{min}$, is about 15 F. In a four-stroke engine the downward fluctuation will be about the same as upward, meaning a temperature range of about 50 F.

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[1] *Forschungsarbeiten*, Heft 263.

The temperature fluctuation does not penetrate deeply; $\frac{3}{8}$ in. from the surface the range of fluctuation is less than 1 F.

Heat Flow.—When the rotative speed of an engine is increased, the duration in seconds of all events, during each cycle, is decreased, but the increased piston speed creates a greater turbulence, slightly increasing U; and as a result the percentage of heat rejected to the jacket slightly increases with the engine speed.

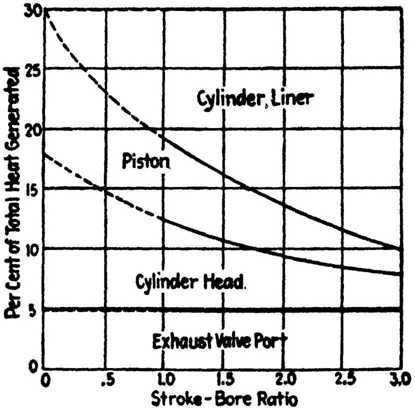

FIG. 21-4.—Heat-loss distribution in four-stroke engines.

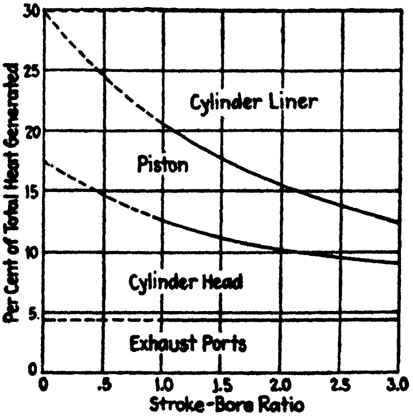

FIG. 21-5.—Heat-loss distribution in two-stroke engines.

Tests have shown that the percentage of jacket loss is nearly independent of the engine load and slightly decreases with an increase of the cylinder diameter.

Since water-cooled engines constitute the majority, they will be discussed first.

Figure 21-4 shows the distribution of heat flow in four-stroke engines of different stroke-bore ratio l/d.[1] The left ends of the curves, for $l/d < 1$, were drawn to show the general direction of the curves. An average figure of 30 per cent of the total heat generated is assumed as the basis for the distribution.

Figure 21-5 gives a similar picture for two-stroke engines.

21-3. Liquid Cooling. *Temperatures.*—Until recently, it was considered good practice to operate all engines so that the water-temperature rise would be small, 10 to 20 F, and to maintain a moderate water-outlet temperature of some 120 to 140 F and not over 160 to 180°, when using an enclosed cooling system. Such low water temperatures were used mainly to reduce the danger of scale formation, which is particularly great with horizontal engines.

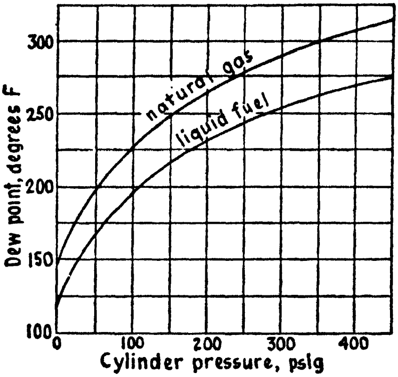

Fig. 21-6.—Condensation of water vapor at different cylinder pressures.

The dew point of the water vapor in the combustion products depends upon the pressure and hydrogen content of the fuel. Figure 21-6 shows that, with the cooling temperatures as given above, a considerable condensation of water is bound to occur on the cylinder walls. The water causes corrosion, which seems to be one of the main causes of cylinder wear.

Tests conducted around 1937 followed up by careful observation of some 200 installations in this country and similar tests in Great Britain have shown that permitting the water temperature to rise above the boiling temperature, to about 220 to 250 F, gives very important and far-reaching advantages:

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[1] Purday, H. F. P., "Diesel Engine Design," 3d ed., p. 170, Constable & Company, Ltd., London, 1928.

1. It eliminates the condensation of the water vapor contained in the products of combustion which in turn:

(*a*) prevents, or at least reduces, materially the washing off of the lubricating oil film from the cylinder and piston-ring surfaces and prevents the formation of sulphuric acid from SO_2, very often contained in the products of combustion; these two factors reduce the wear of the cylinder, piston rings, and valves considerably, under certain conditions, down to one eighth of the usual amount;

(*b*) eliminates crankcase condensation and sludging of the lubrication oil;

(*c*) makes possible the use of "sour" gas and sewer gas which contain acid-forming constituents.

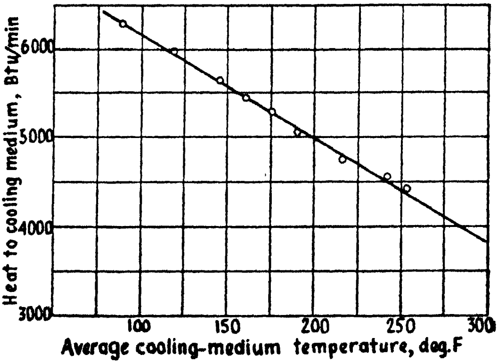

FIG. 21-7.—Effect of cooling-medium temperature on heat rejected to the cooling medium in liquid-cooled engines.

2. It lowers the viscosity of the cylinder-lubricating oil, thus raising the mechanical efficiency of the engine and lowering the specific fuel consumption up to 10 per cent, at full load.

3. It reduces considerably the amount of water which must be circulated, since the cooling effect of each pound of evaporated water is about 970 Btu/lb, instead of 10 to 20 Btu/lb, because of the temperature difference; this increases the fuel saving.

4. It increases the temperature difference between the cooling water and air to which the heat is rejected and, if a radiator is

used, requires a considerably smaller radiator surface and a smaller fan, which again saves fuel.

Quite naturally, with an increase of the jacket temperature, the heat absorbed by the jacket from the gas in the cylinder decreases because of a smaller temperature difference, illustrated by Fig. 21-7 for a particular engine.[1] The heat not transferred to the jacket water increases the exhaust-gas heat and raises their temperature.

Other Liquids.—The use of ethylene glycol, or Prestone, which at atmospheric pressure has a boiling temperature of 387 F, gives the same advantages, except No. 3, if the jacket temperature is maintained at the same level. Its specific heat is 0.675 Btu/lb at 212 F and about 0.775 Btu/lb at the boiling temperature.

The cooling of the pistons of large engines is sometimes done by equipping them with a jacket and circulating oil through it. Pistons of high-speed engines are eventually cooled by a jet of oil directed toward the underside of the piston top.

The use of higher jacket temperatures, up to 250 F, does not require any changes in the construction of the engine or in the lubricating-oil specifications.

Construction Features. *Cylinders.*—In small- and medium-size engines the water jacket is cast together with the cylinder. To reduce the weight, as in aircraft engines, the water jacket is sometimes made from sheet metal. With a steel cylinder a steel jacket may be welded to it. In Beardmore engines the cylinder is of cast iron and the water jacket is of electrodeposited copper; this is formed in place over a wax matrix, which is later melted out. In larger engines the cylinder is formed by a cast-iron liner inserted into a cast-iron jacket.

The details of construction will be discussed later. At this point it will suffice to mention that the water space between the cylinder proper, or liner, and the water jacket should be made such as to ensure a fair velocity of water circulation, at least 5 fpm in stationary engines and up to 60 fpm in light engines.

Cylinder Heads.—In a four-stroke engine the heat carried away by the water that cools the head comes from two places: from the bottom plate which forms one of the combustion space walls, and from the exhaust passage and exhaust valve if the latter is not water cooled. Figure 21-4 shows approximately the relative amounts.

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[1] TAYLOR, C. F., and TAYLOR, E. S., "Internal-combustion Engines," p. 132, International Textbook Company, Scranton, Pa., 1938.

In cylinder heads the problem of good cooling is not so much one of creating sufficient water velocity as of eliminating air and steam pockets, of obtaining, as far as possible, uniform water velocities in all parts of the water space, and of avoiding narrow water passages that are apt to become closed by scale and thus to impair proper circulation.

Exhaust valves need cooling only in large-size engines. Formerly the water was admitted and carried away either through flexible tubing or through telescopic tubes with packing glands.

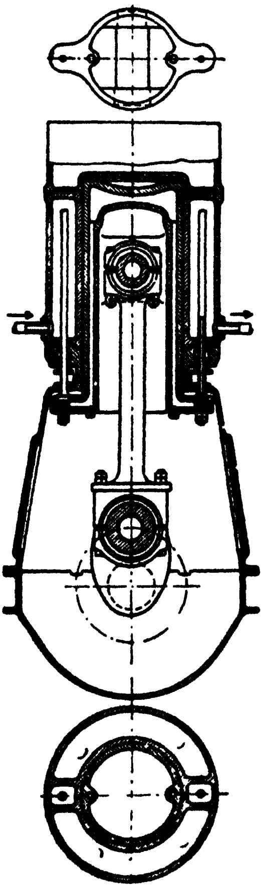

FIG. 21-8.—Piston cooling with water.

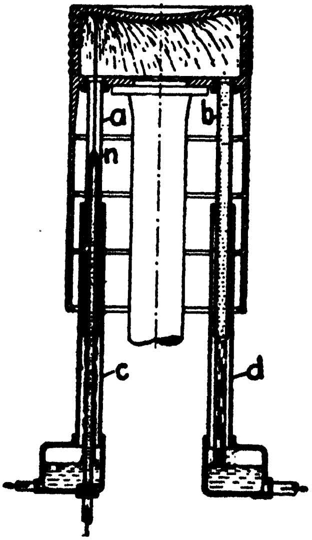

FIG. 21-9.—Jet cooling of piston.

Lately, with the use of heat-resistant steels or special cast-iron valve heads, even large engines are built with uncooled exhaust valves but have cooled valve cages, Fig. 18-10, and effective cooling of the valve seats.

Trunk Pistons.—With very few exceptions single-acting engines have trunk pistons. Trunk pistons dissipate heat to the cylinder walls and lubricating oil very well, so that they do not require special cooling in sizes up to about 22 in. diameter. Formerly, in many engines, cooling of larger pistons was effected by circulating oil. Leakage of cooling oil into the lubricating-oil sump is of little consequence, whereas leakage of water, especially of salt water in marine engines, is likely to cause trouble with bearings.

With the perfection of the design of the water-circulating system many engines now use water for piston cooling. In the older system, still used by some manufacturers, water is admitted through telescopic tubes with glands and circulates under pressure of 40 to 60 psi, Fig. 21-8. It has been found that cooling is more efficient if the water does not fill the whole inner space and can splash in it, Fig. 21-9. The water is delivered under pressure to nozzle *n*, is injected into the piston, and returns by gravity. Tubes *c* and *d* do not touch tubes *a* and *b* but do prevent the splashing of water into the crankcase.

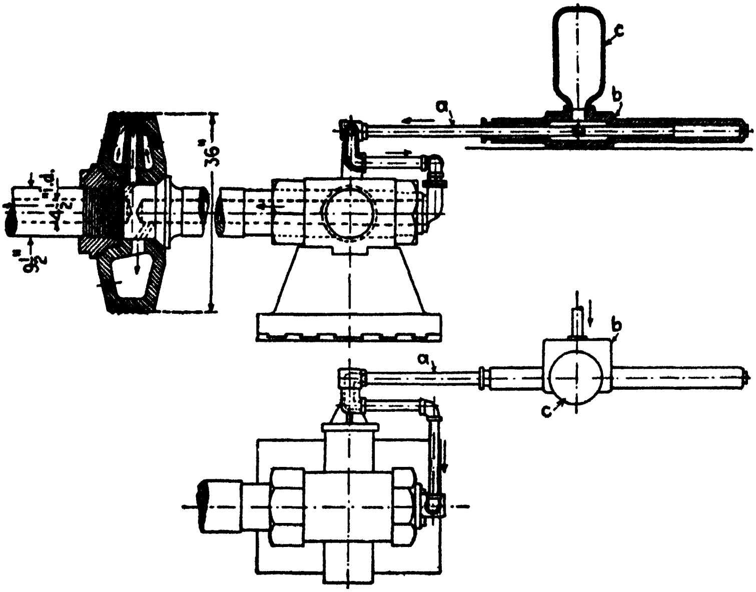

FIG. 21-10.—Double-acting gas-engine piston with telescope-pipe cooling.

Barrel Pistons.—Some single-acting and all double-acting engines are built with crossheads and barrel pistons. The cooling water for the latter is delivered and discharged through the crossheads and hollow piston rods by one of the following schemes: the water connections to the crosshead are made by means of telescopic reciprocating tubes *a* and glands, Fig. 21-10, or by means of telescopic tubes and swivel joints, Fig. 27-12, or, finally, by using a discharge nozzle, Fig. 27-13, which creates enough velocity to raise the water and force it into the upper point from which it is allowed to flow away by gravity as in trunk-piston cooling, Fig. 21-9.

With reciprocating telescoping tubes an air chamber *c*, Fig. 21-10, is necessary to absorb the hydraulic shocks resulting from the variation of the volume.

Piston rods in crosshead engines must be cooled, but this is obtained when the cooling water is admitted through the crossheads to the piston.

21-4. Water Circulation. *Quantity.*—The quantity of water that must be circulated depends upon the initial temperature and the desired temperature rise of the water. The initial temperature depends upon the atmospheric conditions, either directly as in marine engines, or indirectly if a recooling system is used and the water is circulated over and over. In order to avoid excessive heat stresses, the temperature difference between the incoming and outgoing water must not exceed 20 F in small- and medium-size engines and less, down to 10 F, in large engines.

The final water temperature in large engines usually was not allowed to go above 120 F. For engines of medium size a maximum temperature of 130 to 140 F was allowed. In automobile engines the cooling water may reach the boiling point without any damage to the engine, but 180 F was considered as the desirable limit. The investigations concerning cooling by evaporation, with jacket temperatures ranging from 215 to 250 F, mentioned above, promise to change the situation. If the engine is cooled by untreated water, which always contains foreign matter, the temperature should be kept low enough to prevent precipitation of the impurities and formation of scale. In an engine using salt water the temperature should not exceed 110 to 115 F.

The water is usually circulated first through the cylinder jacket, then the cylinder head, and from there to the exhaust-valve cage. Pistons are cooled usually from a separate pipe line.

The quantity of water to be circulated in gallons per hour is

$$G = Q/(t_2 - t_1)8.32 \tag{21-12}$$

For average conditions, the heat flow to the water jacket is about 3000 Btu/hp-hr for large engines, increasing to about 3500 and up to 4000 Btu/hp-hr for small and less efficient engines.

Excessive water circulation resulting in low final water temperature is not desirable, particularly in gasoline engines, as it will cause a precipitation of a considerable portion of the fuel, disturb the charge distribution, increase the fuel consumption, and decrease the useful power.

A low cooling-water temperature increases the viscosity of the lubricating oil and consequently the piston friction. The difference between friction loss at high and low jacket temperature may amount to as much as 8 per cent of the indicated horse power if the piston is large and heavy and drops to about 4 per cent if the piston has small bearing area and weight. See also Fig. 16-3.

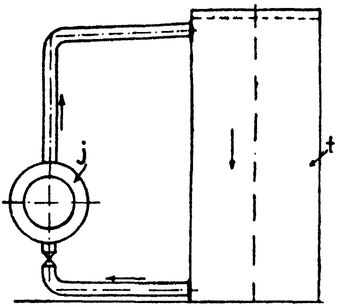

FIG. 21-11.—Gravity-circulation cooling.

Circulation.—Two methods of circulation are in use—gravity circulation and forced circulation. Gravity circulation, also called *thermosiphon circulation*, is based on the fact that when water is heated its density decreases and it tends to rise, the colder particles sinking to take the place of the rising ones. Circulation is obtained if the water is heated at one point and cooled at another. Gravity circulation is used in small engines only—seldom over 30 hp.

Figure 21-11 shows the gravity-circulation arrangement for a small horizontal engine. Water heated in the cylinder jacket *j* flows to a tank *t* where it is cooled by radiation, gradually descends to the bottom, and flows back to the engine. Figure 21-12 shows gravity circulation as applied to an automobile engine. To obtain proper water circulation the connections between the engine jacket *j* and the radiator *t* must present small resistance to the water flow and be wide, short, and have as few bends as possible. Even under favorable conditions circulation is slow, especially when the temperature difference is small, as at light loads. At heavy loads the jacket heat may exceed the heat radiated by the cooling element and the water in the jacket is apt to boil. This system is used only in smaller engines where simplicity is of importance.

Most engines have forced circulation by pumps, of either the centrifugal or plunger type. The advantage of the forced circulation is the ease of controlling the jacket-water temperature. This may be accomplished either by regulating the opening of the valve between the pump and the engine, or by regulating the water-discharge valve of individual cylinders. Figure 21-13 shows a pump circulation system in an automobile engine with a centrifugal pump *p* and automatic temperature control by a by-pass *b* with valve *v* operated by a thermostatic element *e*.

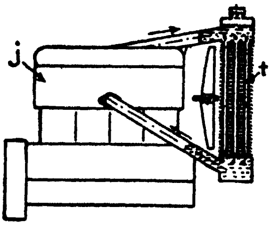

FIG. 21-12.—Thermosiphon circulation.

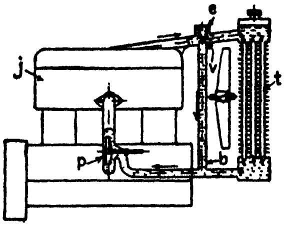

FIG. 21-13.—Forced circulation with by-pass.

Evaporative Cooling.—If the water in the cylinder jacket is allowed to boil, 1 lb evaporated water will absorb heat equal to the latent heat of vaporization or about 970 Btu. This is from 24 to 48 times more than the heat carried away by 1 lb circulating water with a temperature rise of 20 to 40 F. Neither pump nor radiator being required, this system has the advantage of simplicity and is used for small stationary and tractor engines. The water jacket is made large at the top, forming a so-called *hopper*. The quantity of water in the hopper must be sufficient to run the engine for several hours without adding water. The evaporative system is not advantageous if the water contains impurities which form scale on the cylinder walls.

21-5. Recooling of Water.—Only seldom, as in small marine engines, is the cooling water used once and discarded. The recooling of water for continuous use can be effected by one of the following means:

(*a*) direct evaporation;
(*b*) heat exchangers with a secondary water circulation;
(*c*) radiators with atmospheric air as a coolant.

By the first method, called an *open cooling system*, the water from the jacket is discharged either into a cooling pond or to the top of a cooling tower and is cooled by the latent heat of evaporation of the part carried away by the air. The advantage of this system is its simplicity and a small expenditure of energy for circulation of the water. Its big drawback is a gradual contamination of the water by salts as pure water is evaporated and the make-up water added always contains some salts. As the salt concentration reaches a certain limit, all water must be drained and fresh water put into the system. However, even if this is done regularly, a certain amount of sediment is deposited in the engine jackets and forms scale, which eventually causes cracks, usually in the cylinder head. At the same time, this system requires low jacket temperatures, with the ensuing drawbacks mentioned before.

A closed system uses either distilled or treated soft water which is passed through a heat exchanger, where it is cooled and then led back to the jacket. The heat exchanger may be either simply a coil in the basin of a cooling tower or a shell and tube exchanger. In the latter the jacket water passes through the tubes and the cooling medium through the shell. In oil-pipeline pumping stations, the pumped oil is used as the coolant. A closed system permits the use of any jacket temperature up to the highest desirable; and if the amount circulated is large enough, the temperature difference between the incoming and outgoing water can be kept low, 10 to 20 F.

The only drawback of the closed system is a slightly greater energy requirement for the two pumps and a higher initial cost. However, the elimination of scale and the advantages of higher jacket temperatures are so important that the use of the closed system has become almost universal.

Radiators.—In a radiator the heat is rejected to the air which passes through it. In automobile engines the air is sucked through by a fan, assisted by the movement of the car; the velocity of an airplane is sufficient to produce the necessary air stream through the radiator. From the previous discussion of heat transfer between water and liquid divided by a metal wall, it is clear that the performance of a radiator depends upon the velocities of air and water. The increase of air velocity, in the first place, increases the quantity of air passing the radiator fins, and, in the second place, by removing the inert air film sticking to the metal surface, it increases the surface-heat-transfer coefficient h_2 according to expression (21-4). The water velocity acts in the same way and increases the surface coefficient h_2, expression (21-5). Since the specific heat of air is less than one-fourth that of water and the surface coefficient between metal and air is many times lower than between metal and water, the air-cooled surface should be considerably greater than the inner surface in contact with the water. This is obtained by adding thin metal fins to the water tubes *t*, Fig. 21-12 or 21-13, which form the water passages between the upper and lower radiator tanks.

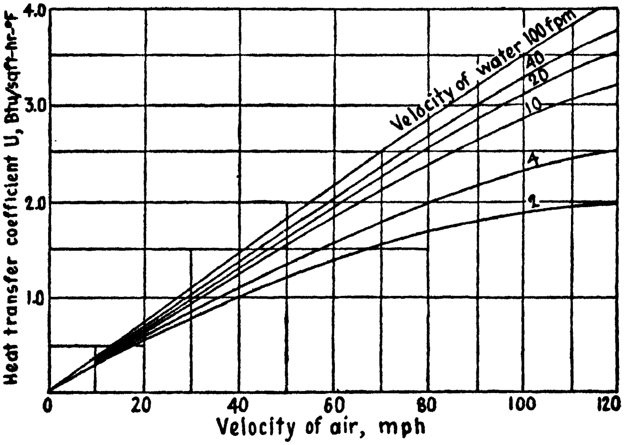

FIG. 21-14.—Performance curves of a radiator.

Figure 21-14 gives performance curves of a typical automobile radiator, the figures over the curves indicating the respective velocities of the water in feet per minute.

In automobile radiators the tubes with fins occupy from 15 to 45 per cent of the total frontal area and present a considerable resistance to the movement of the car. The power absorbed by this resistance is approximately proportional to the cube of the velocity. It becomes very considerable for airplanes with speeds of 100 mph and up. The use of Prestone, which has a boiling point of 387 F as compared with 212 F for water, permits an increase to more than twice the temperature head $t_2 - t_1$ thus more than doubling the heat exchange. The increased temperature head allows a decrease in the same proportion of the heat-exchange area A and consequently of the size, weight, and frontal resistance of the radiator. All this is of a great advantage in an air plane design.

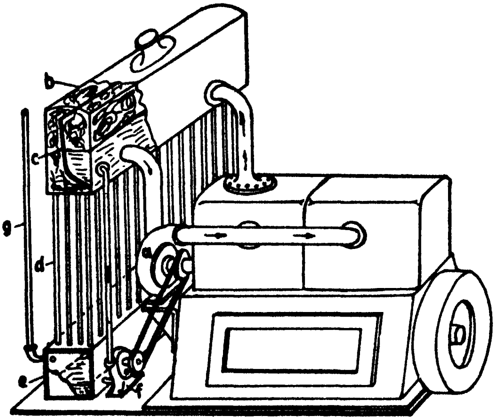

FIG. 21-15.—Vapor-phase cooling with air-cooled condenser.

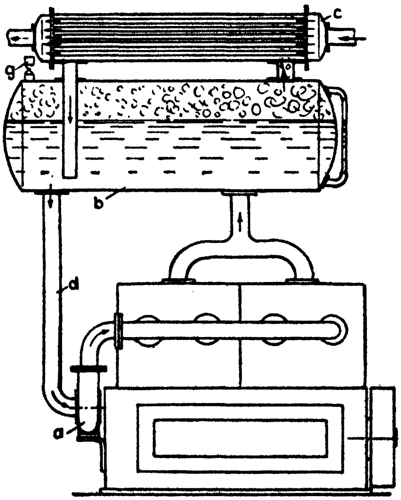

FIG. 21-16.—Vapor-phase cooling with water-cooled condenser.

Vapor-phase Cooling.—The advantages of high-temperature jackets enumerated in Sec. 21-3 apply particularly to a system developed in this country, called *vapor-phase cooling,* Fig. 21-15. The water is circulated by the pump a and when delivered to the over-head tank b, part of it boils out. The vapor rises over the partition c and, because of the condensing action of the radiator tubes d, flows down into the lower tank e, from which the condensate is picked up and returned to the tank b by the small pump *f*. The vertical pipe *g* is a communication with the outside atmosphere to prevent the collapsing of the tanks *b* and *e* when the pressure inside them, due to condensation, falls below atmospheric.

For larger engines the condensation of the vapor formed in the overhead tank *b*, Fig. 21-16, occurs in the heat exchanger *c*, cooled by a secondary water circuit, and the water returns to *b* by gravity.

21-6. Air Cooling.—Because of the low value of the heat-transfer coefficient *h* between metal and air, the wall temperature of air-cooled cylinders is considerably higher than that of the water-cooled type. In order to lower the cylinder-wall temperature, the outside surface must be increased by fins. Experiments have shown that for satisfactory operation the cylinder-head temperature of most engines should not exceed 570 to 600 F. However, in the Franklin automobile engines the temperature of the cylinder head near the exhaust valve reached 650 to 700 F, without ill effects.

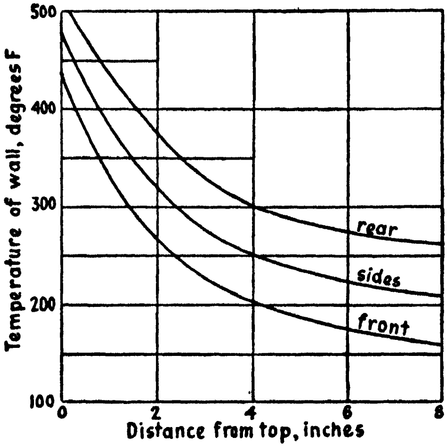

FIG. 21-17.—Temperature distribution in the surface of an air-cooled cylinder.

Figure 21-17 shows typical temperature variations as observed on the outside surface of a finned air-cooled cylinder, from the top of the cylinder to the other end and at different points in respect to the air stream.

Heat Dissipation.—The amount of heat which must be dissipated by an air-cooled cylinder at full load is from 1500 to 2300

Btu/hp-hr; in addition, the lubricating oil gives up in the oil cooler 100 to 1200 Btu/hp-hr, and a total of from 2000 to 2500 Btu/hp-hr is extracted from the cylinder. Assuming a mean temperature of the cooling surface of 450 F, an air temperature of 70 F, and a surface coefficient $h = 20$ gives for the necessary minimum cooling area

$$A = 1500/[(450 - 70) \times 20] = 0.20 \text{ sq ft/hp}$$

With a lower h resulting from lower velocities, or with a greater heat to dissipate, the necessary area may go up to 0.35 sq ft/hp.

Proper cowling of a radial engine gives more effective cooling with a decreased head resistance. Engines with in-line and V-cylinders require a special hood with baffles to direct the air stream over all cylinders. In the Franklin automobile engines the cylinders had lengthwise fins and air jackets; air delivered by a blower to the cylinder head passed downward through each jacket.

Design Points.—In order to keep down the cylinder-wall temperature, the inner surfaces must be shaped and finished so as to absorb a minimum amount of heat, and the outer surfaces so as to dissipate this heat most readily. A polished metal surface absorbs less heat than a rough surface. Therefore the inner surface, even though it is not in contact with the piston, must be highly polished, and in order to machine the inside of the cylinder head the overhead valve construction is most frequently used. A polished surface has an additional advantage in that carbon formed from burned lubricating oil does not adhere so tightly to the walls and is more readily blown out with the exhaust. The overhead-valve arrangement also gives the minimum combustion-space surface, which decreases the heat given up by the gases. Quick exhausting of the incandescent products of combustion at the end of the expansion stroke is also important, and quick-opening large-size exhaust valves must be used.

The hottest parts of a cylinder head are the exhaust valve and port, and particular attention must be given to their cooling. Contrary to expectation the rear side of a cylinder loses at least as much heat to the air that streams around it as the front side directly impinged on by the air. In radial engines the exhaust valves should be located on the right side of the cylinder looking upon the forward end of the engine along the path of the air thrown back by the propeller.

The following features help to keep down the temperature of the exhaust valve: a heavy head to allow rapid travel of heat from the center to the seat, a wide seat to reject the heat, a large stem, and a heavy valve guide extending fairly close to the head. A hollow valve stem partly filled with sodium, Fig. 18-13, is almost standard practice.

Cooling Fins.—The necessary increase of the outside cooling surface is obtained by thin metal fins placed in the direction of the air stream. These fins are either cast as a part of the aluminum cylinder or turned from the forging in forged-steel cylinders

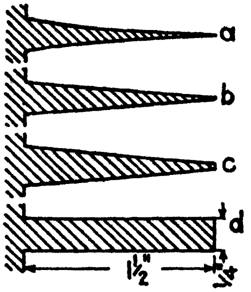

FIG. 21-18.—Sections of cooling fins.

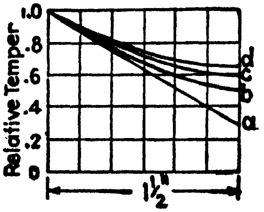

FIG. 21-19.—Relative temperatures of fins.

The heat-dissipating capacity of fins depends upon their cross section and length. The mathematical treatment of this problem is rather involved. The basis for the design of fins can be taken from experimental data.[1] The heat given off to the air is directly proportional to the temperature difference. At the same time as heat is gradually dissipated from the fin surface, the temperature of the fin decreases from its root to its tip. Hence, the fin surface nearer to the tip dissipates heat at a lower rate and is less efficient. On the other hand, as the quantity of heat flowing toward the tip gradually decreases, the thickness of the fin may decrease. The material of a fin is used most efficiently if the drop in temperature from the root to the tip is constant per unit of length. A comparison of fins of different cross sections is given in Fig. 21-18, and the drop in temperature from root to tip in each type is shown by curves in Fig. 21-19.

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[1] DICKSEE, C. B., *Automotive Industries*, Apr. 5, 1923.

The rectangular section *d* has the smallest temperature drop, dissipates less heat, and being heavier, represents rather inefficient use of material; such a shape is not used.

Cast fins are made with a total taper of from 3½ to 5°, or about 0.06 to 0.09 in. per inch length, to give sufficient draft to the pattern. The tip is made 0.02 to 0.05 in. thick and a clearance of 0.10 to 0.20 in. is allowed at the root, spacing the fins 4 to 7½ to the inch. The fins are made from 1 to 1½ in. long. Fins turned out of a solid steel forging are made with a smaller taper, about 0.02 in. per inch length.

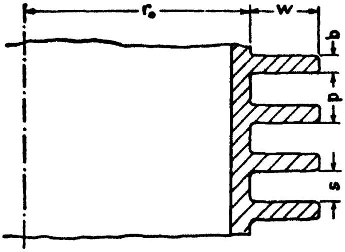

FIG. 21-20.—Fin dimensions for use in equation (21-13).

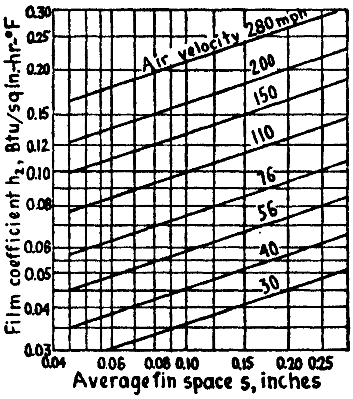

FIG. 21-21.—Relation between fin spacing, film coefficient, and air velocity.

Fin thickness b, Fig. 21-20, for a given space *s* does not have an appreciable effect upon heat dissipation.[1]

Fin Spacing.—For closely spaced fins the mutual interference of the boundary layers of adjacent fins restricts the air flow and results in a small film coefficient h_2. As the fin spacing is increased, h_2 increases until a point is reached where the adjacent fins do not interfere any more. Curves representing an average of the surface coefficients determined in such tests are shown in Fig. 21-21.[1]

Fin Width.—Tests with fins of different width *w* have shown that for the width as used in practice, ½ to 1½ in., h_2 can be assumed to be independent of *w*.

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[1] BIERMAM, A., and PINKEL, B., Heat transfer from Finned Metal Cylinders in an Air Stream, *NACA Rept.* 488, 1934, p. 10.

Heat Flow.—The computations of the heat Q dissipated by a cylinder with fins are much simplified by the use of an equation of a structure similar to the general equation, namely

$$Q = A_o U'(t_3 - t_4) \qquad (21\text{-}13)$$

where A_o is the outside surface of the cylinder, sq ft,

t_3 is the average temperature of this surface, °F,

t_4 is the temperature of the cooling air, °F,

U' is a special heat-transfer coefficient, Btu/sq ft-hr-°F

The expression for this special heat-transfer coefficient is

$$U' = \frac{h_2}{s + b}\left\{\frac{2}{a}\left(1 + \frac{w}{2r_0}\right)\tanh\left[a(w + 0.5b)\right] + s_0\right\} \qquad (21\text{-}14)$$

where s, b, r_0 and w, Fig. 21-20, in.,

s_0 is the spacing of tapered fins on the cylinder surface, in.,

h_2 is the film coefficient, Btu/sq ft-hr-°F, values from Fig. 21-21 multiplied by 144,

a is a factor,

$$a = \sqrt{2h_2/kb} \qquad (21\text{-}15)$$

k is conductivity, Table 21-1, Btu/hr-sq ft-°F-in.

Altitude.—The change of density of the air must be taken into account by using a corrected air velocity v', mph. Figure 21-21 is plotted for air of 80 F and 29.92 in. Hg., $w_0 = 0.0734$ lb/cu ft; therefore

$$v' = vw_a/0.0734 \qquad (21\text{-}16)$$

A similar correction must be introduced when using expression (21-4), in which the constants refer to standard conditions.

Example 21-3.—Detemine the heat dissipated per square foot of the outside cylinder surface of a 6-in. cylinder; the cylinder-wall thickness is 0.20 in.; the other dimensions are: at the tip $b = 0.04$ in., $w = 1.5$ in., the taper is 0.02 in./in.; the average cylinder temperature $t_3 = 325$ F, the air velocity is 200 mph, the engine operates at an altitude of 20,000 ft; six fins per in.

The fin thickness at the root is $b_0 = 0.04 + 1.5 \times 0.02 = 0.07$ in.; the average thickness $b = 0.04 + 0.015 = 0.055$ in.; the spacing at the root $s_0 = 1/6 - 0.07 = 0.097$ in., and the average spacing

$$s = 1/6 - 0.055 = 0.112 \text{ in.}$$

At a 20,000-ft altitude the pressure, according to equation (7-11),

$$p = 14.7/\text{antilog}_{10}\,(20{,}000/122.86I_m) = 14.7/2.18 = 6.75 \text{ psia}$$

The temperature, by expression (7-12),

$$t = 60 - 0.0036 \times 20{,}000 = 60 - 72 = -12 \text{ F or } 448 \text{ R}$$

The air density under these conditions, from the characteristic equation,

$$w_a = 6.75 \times 144/(53.35 \times 448) = 0.0407 \text{ lb/cu ft}$$

and the corrected air velocity, by expression (21-16),

$$v' = 200 \times 0.0407/0.0734 = 110.9 \text{ mph.}$$

From Fig. 21-21 for $s = 0.112$ in. and $v' = 110.9$ mph,

$$h_2 = 0.101 \times 144 = 14.5 \text{ Btu/sq ft-hr-°F.}$$

By expression (21-15), with $k = 310$, Table 21-1,

$$a = \sqrt{2 \times 14.5/(310 \times 0.055)} = 1.304$$

By expression (21-14)

$$U' = \frac{14.5}{0.112 + 0.055}\left\{\frac{2}{1.304}\left[1 + \frac{1.5}{6 + 2 \times 0.20}\right] \tanh [1.304(1.5 + 0.5 \times 0.055)] + 0.097\right\} = 166.4 \text{ Btu/hr}$$

And by equation (21-13) the heat dissipated by 1 sq ft

$$Q = 1 \times 166.4(325 - (-)12) = 56{,}100 \text{ Btu/hr.}$$

Baffles.—The film coefficient can be considerably increased by using baffles, which force the air through the space between the fins. Figure 21-22 shows the various types of baffles that may be used.[1] The baffling of Fig. 21-22*a* has a large pressure drop $p_1 - p_2$. The ideal case would be a negligible kinetic energy at the entrance and exit points, *KE'* and *KE''*, with a well-rounded

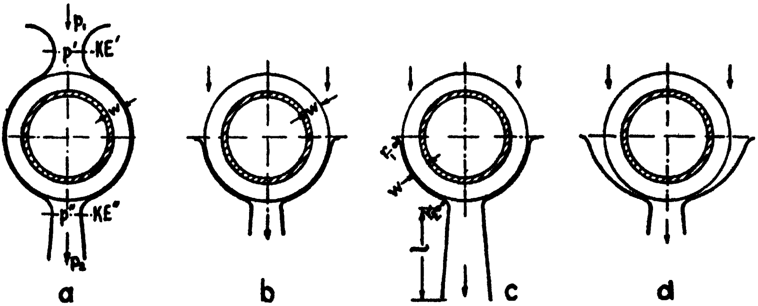

Fig. 21-22.—Baffle types for air-cooled cylinders.

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[1] Brevoort, M. J., *NACA Tech. Notes* 649, 1937.

entrance to reduce the entrance loss and an exit section that will transform the velocity head into pressure head and thus decrease the pressure drop $p_1 - p_2$, Fig. 21-22*c*. However, Fig. 21-22*b* shows the normal type of baffiing. Baffiing by Fig. 21-22*d* has a smaller resistance but is less effective. The influence of baffling in connection with the width and spacing of the fins and air-pressure drop can be computed on the basis of published investigations,[1] but the procedure is ráther complicated and the results not sufficiently definite to be dealt with here. Fig. 21-23*a* shows a poor head baffle and Fig. 21-23*b* a good one.

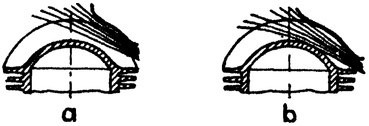

Fig. 21-23.—Cylinder-head baffles.

Figure 21-24 shows a polar diagram of temperatures around a 4.66-in. finned engine cylinder: curve *a* for a bare cylinder and *b* for a cylinder with a normal baffle.[2] Baffling has reduced the maximum temperature from about 300 to 200 F, and what is still more important has reduced the temperature difference between front and rear from

$$300 - 185 = 115 \text{ F}$$

to

$$200 - 150 = 50 \text{ F}$$

which means smaller stresses and less distortion.

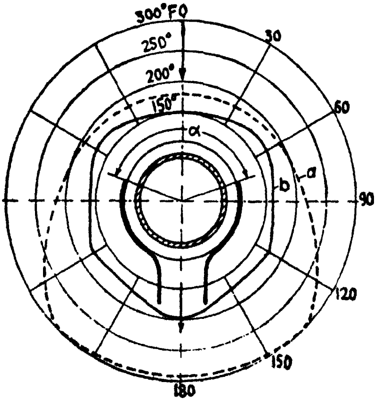

Fig. 21-24.—Temperature distribution around a finned cylinder with and without baffles.

21-7. Temperature Effects. Many problems in the design of internal-combustion engines are closely connected with the effects caused by the temperatures of different parts of the engine. These effects are:

1. Expansion of metals.
2. Stresses caused by temperature differences in metal parts.
3. Decrease in strength of metals at higher temperatures.

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[1] Bierman, A. E., The Design of Metal Fins for Air-cooled Engines, *SAE Journal*, vol. 41, p. 388, 1937; Brevoort, M. J., *NACA Tech. Notes* 620, 1937, and 655, 1938.

[2] Kemper, C., *SAE Journal*, vol. 35, p. 375, 1934.

4. Deterioration of metal surfaces when subjected to high and varying temperatures.

5. Growth of cast iron.

All these effects must be considered in a correct design and in many instances can be controlled by proper cooling. These temperature effects set the limits to the cylinder sizes of reciprocating engines and also present the main difficulties in the development of a practical internal-combustion turbine.

Expansion.—The coefficients of linear expansion of metals within the range of temperatures occurring in internal-combustion engines can be considered constant. Hence, the linear expansion is directly proportional to the temperature increase, and, per unit length, it is equal to

$$\Delta l = a(t_2 - t_1) \tag{21-17}$$

The coefficients of expansion vary slightly with the temperature and the composition of alloys, but in most cases average values, as given in Table 21-1, may be used.

Coefficients of expansion of special alloy steels used for exhaust valves should be obtained from the steel manufacturers.

Temperature Stresses.—If a metal bar is rigidly fixed so that it cannot expand with the change of temperature from t_1 to t_2, the material will be stressed as though it were compressed Δl in.

The resulting stress per square inch can be calculated from Hooke's relation

$$s = E\Delta l = Ea(t_2 - t_1) \tag{21-18}$$

where E is Young's modulus of elasticity, Table 21-1.

In the case of a wall in which the temperature gradually changes from t_1 on one side to t_2 on the other, there will be a neutral line about halfway between the inner and outer surfaces. The outer, colder fibers will stretch and take half of the load, decreasing by one-half the compression of the inner fibers. Thus the equation for the stress, taking into account the transverse deformations,[1] becomes

$$s = E\Delta l/2(1 - \mu) = Ea(t_2 - t_1)/2(1 - \mu) \tag{21-19}$$

where μ is Poisson's ratio, Table 21-1.

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[1] TIMOSHENKO, S. P., "Theory of Elasticity," p. 210, McGraw-Hill Book Company, Inc., New York, 1934.

The fibers with the higher temperatures will be subjected to compressive stresses; the fibers with the lower temperatures, to tensile stresses.

For the wall of a cylinder, designating $D_o/D_i = 1 + m$, where D_o is the outside diameter and D_i the inside diameter, the stress on the outside surface is[1]

$$s_o = aE(t_i - t_o)(1 - m/3)/2(1 - \mu) \qquad (21\text{-}20)$$

and the stress at the inside surface

$$s_i = -aE(t_i - t_o)(1 + m/3)/2(1 - \mu) \qquad (21\text{-}21)$$

If $t_i > t_o$, as in a cylinder liner, s_o will be positive, or a tensile stress, and s_i a compressive stress. If $t_i < t_o$, as in a water-cooled hollow piston rod, s_o will be a compressive stress and s_i a tensile stress.

Under certain conditions these stresses may exceed the elastic limit in certain points, and destruction will start at first in the shape of minute cracks, which gradually increase in number and size. As these cracks go deeper, the strength of the part is impaired until its failure occurs.

Presenting the temperature drop $t_i - t_o$, similar to equation (21-9), as

$$t_i - t_o = qb/k \qquad (21\text{-}22)$$

where q is the heat flow, Btu/sq ft-hr, and in the case of a cylindrical wall is referred to the mean area. The expressions for the stresses now can be presented as

$$s_o = aEqb(1 - m/3)/2k(1 - \mu) \qquad (21\text{-}23)$$

and

$$s_i = -aEqb(1 + m/3)/2k(1 - \mu) \qquad (21\text{-}24)$$

These equations show that for a certain material, constant a, E, k, and μ, the stresses are directly proportionate to the heat flow q and wall thickness b. If the stress is too high and b cannot be reduced because of other requirements, the best way to overcome the difficulty is to use a different material with a higher allowable stress and hence a smaller thickness b.

Decrease of Strength.—With an increase of the temperature, the tensile strength of most alloys used in internal-combustion engines slightly increases up to 300 to 400 F; after that it begins to decrease. The rate of decrease depends to a great extent upon the composition of the alloy. However, the more important elastic limits decrease with the temperature, at first slowly, and after 500 to 600 F very fast, Fig. 21-25.

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[1] TIMOSHENKO, *op. cit.*, p. 373.

Growth of Cast Iron.—When cast iron is heated, and especially when heated repeatedly, its specific volume begins to increase. Some observations indicate that it is dangerous to heat cast iron

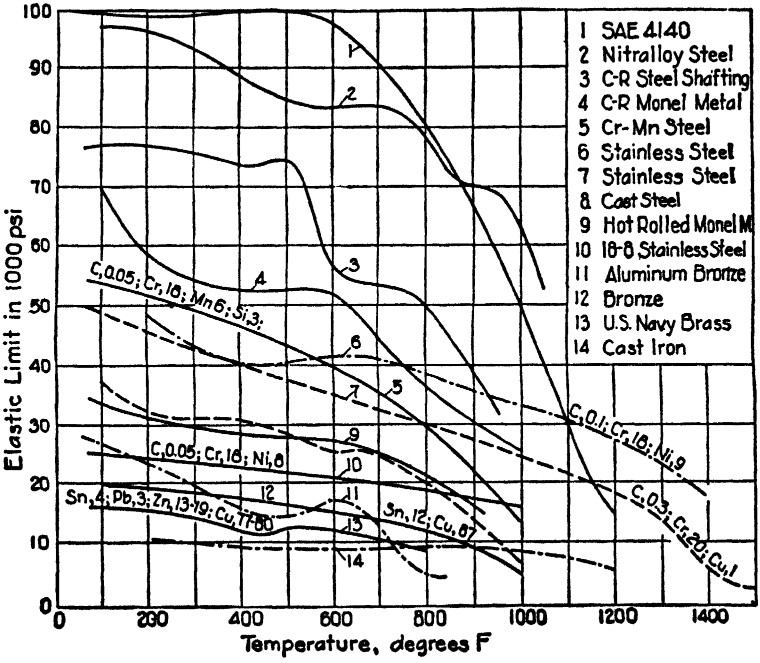

FIG. 21-25.—Temperature influence on elastic limits of metals.

above 550 F. From other observations the limit seems to be about 1100 F. The growth may be due to several causes:

1. Conversion of combined carbon to free carbon.

2. Certain action of the silicon and phosphorus contained in the cast iron.

3. Expansion of minute particles of gases dissolved in the metal when the elastic limit decreases with the temperature.

4. In the case of pistons and cylinder liners of internal-combustion engines, penetration of oil and gas under pressure into graphite particles of cast iron.

This growth is dangerous because it creates local compressive stresses which may result in failure of the part, similar to failure from excessive heat stresses. Also, as in the case of pistons, it may reduce the clearance between the working parts and result in excessive wear or even seizure through friction and lack of lubrication.

The simplest and surest way to eliminate the growth of cast iron regardless of its cause has been found in the addition of steel to cast iron. A suitable composition of such semisteel castings is, in per cent: 2.5 to 3.25 C, 0.5 to 1.5 Mn, 0.6 to 1.2 Si; 0 to 0.9 Ph; 0 to 1.0 S. Addition of a small amount of nickel, about 1 per cent, also checks this phenomenon of growth.

21-8. Problems.—**1.** Determine the heat transfer, through the cylinder only, of a $4\frac{3}{4}$-in. × 6-in. gasoline engine with a compression ratio $r = 5.75$ operating at 1800 rpm. The cast-iron cylinder wall is $\frac{5}{16}$ in. thick. Assume a mean gas temperature of 900 F. The cooling medium is air of 70 F and a velocity of 80 mph.

2. Determine the inside and outside cylinder-wall temperatures for the foregoing problem.

3. Find how much the wall temperatures in problem 2 will be affected if instead of cast iron the wall is made of a $\frac{1}{8}$-in. thick steel liner shrunk into a $\frac{3}{16}$-in. aluminum jacket.

4. Determine the additional outside surface required to reduce the inside wall temperature in problem 2 to 450 F. How much additional heat will be transferred in this case?

5. Determine the inside and outside cylinder-wall temperatures for the engine of problem 1, if the cooling medium is water with a mean temperature of 140 F, flowing through the jacket with a velocity of 6 fpm.

6. Determine how many gallons of water must be circulated per minute in problem 5, if the water is admitted at a temperature of 125 F.

7. Determine the approximate distribution of jacket loss in the engine of problem 5 between the exhaust valve, cylinder head, piston, and liner.

8. Determine the heat transfer per hour to the cooling water of a three-cylinder $10\frac{1}{2}$-in. × 12-in. four-stroke gas engine. The compression ratio is 5.5:1; the engine speed is 257 rpm; the cast-iron cylinder and cylinder-head walls are $\frac{15}{16}$ in. thick. Assume a mean gas temperature of 815 F; the water velocity is 2.5 fpm; the water is admitted at 140 F, and a temperature rise of 25 F is allowed.

9. Check the temperature rise of 25 F in problem 8, with the water velocity of 2.5 fpm.

10. Determine the inside and outside cylinder-wall temperatures in problem 8.

11. Determine the approximate distribution of jacket loss in the engine of problem 8 between the exhaust valve, cylinder head, piston, and liner.

12. Compute how many gallons of cooling water must be circulated per minute in the engine of problem 8.

13. Determine the heat transferred to the jacket by the engine of problem 8 if a vapor-phase cooling system is used with a pressure of 5 psig in the jacket space.

14. Determine the necessary capacities of the two pumps, Fig. 21-15, for the engine of problems 8 and 13.

15. Determine the amount of water to be circulated for the vapor-phase cooling of the engine as given in problem 13, using the scheme of Fig. 21-16.

16. Determine the size of a sheet-metal tank, Fig. 21-11, for a 10-hp gas engine using 12 cu ft/hp-hr of gas with a heat value of 1050 Btu/cu ft. Jacket loss is 30 per cent. Find the minimum sizes of the connecting pipes.

17. Determine the size of a sheet-metal tank, Fig. 21-11, for a 6-hp oil engine using 0.50 lb/hp-hr of fuel oil of 30° API. The jacket loss is 28 per cent. Find the minimum sizes of connecting pipes.

18. Determine the necessary size of an evaporative-cooling hopper for a 5-bhp engine using 0.6 lb/hp-hr fuel oil. The heat value of oil is 18,500 Btu/lb. The jacket loss is 28 per cent. The hopper must be refilled not oftener than every 5 hr.

19. Find the heat dissipated by a $3\frac{1}{2}$-in. × $3\frac{3}{4}$-in. cylinder with turned steel fins having a width of $\frac{3}{4}$-in., a thickness at the root of $\frac{1}{16}$ in., and a taper of 0.02 in. per inch width, 5 fins per inch. The cylinder wall is $\frac{1}{8}$ in. thick, its temperature is 420 F, air 75 F; air velocity is 175 mph.

20. Determine the thickness of the air film participating in carrying away the heat from the fin in problem 19, if the air temperature rise is 35 F.

21. A $4\frac{1}{4}$-in. × $3\frac{3}{4}$-in. × 1900 rpm aircraft-engine cylinder has a 0.14-in. steel wall and turned steel fins; the average $w = 0.825$ in., $b = 0.06$ in., $p = \frac{1}{4}$ in., the average outside wall temperature is 300 F; the mean effective pressure is 100 psi; the air velocity is 95 mph. Find the heat dissipated per hour at (*a*) sea level and (*b*) at an altitude of 10,000 ft.

22. For the engine of problem 21 find the temperatures of the inside cylinder surface and the probable mean gas temperature.

23. Work problem 21 assuming that the cylinder has a 0.10-in. steel liner and a 0.12 in. aluminum shrunk-on jacket with cast fins; the same number, width, and tip thickness of fin is used, but with a taper of 0.06 in./in., the outside wall temperature is 263 F.

24. For the engine of problem 23 find the temperature of the inside cylinder surface and the probable mean gas temperature.

25. A $6\frac{1}{8}$-in. × $6\frac{7}{8}$-in. aircraft engine cylinder has a 0.20-in. steel wall and turned steel fins; the compression ratio is 6.40; $w = 1$ in., $b = 0.05$ in., no taper, 7 fins per inch; the average outside wall temperature is 330 F; the bmep is 143 psi at 2100 rpm. The air velocity is (*a*) 150 mph at sea level and (*b*) 250 at an altitude of 25,000 ft; find the heat dissipated per hour per cylinder for both conditions.

26. Having made necessary additional assumptions and computed additional data, find the amount of heat dissipated by the cylinder head and piston of the engine of problem 25 and check the percentages assuming a gasoline consumption of 0.51 lb/hp-hr; the head is of aluminum, $\frac{5}{8}$-in. thick inside, has a practically hemispheric surface and has cast fins, 4½ fins per inch; the average width is 1¾ in., thickness $b = 0.03$ in., taper 0.05 in./in.

27. Determine the axial expansion of a hollow piston rod 14 ft 6 in. long, 8¼ in. outside and 4½ in. inside diameter, if the water flowing through the rod has a mean temperature of 170 F and the outside surface of the rod has an average temperature of 420 F; the heat flow is 18,000 Btu/sq ft-hr.

28. Find the temperature stresses in the piston rod of problem 27 taking into consideration both the axial and radial expansions.

29. Determine the temperature stress in a cast-iron 15-in. cylinder liner, 1¼ in. thick with a maximum heat flow of $q = 45{,}000$ Btu/sq ft-hr; also find the inside temperature of the liner if the outside temperature is 170 F.

30. Find the temperature stress in a 4½-in. steel liner $\frac{3}{16}$ in. thick with a maximum heat flow of 30,000 Btu/sq ft-hr.

CHAPTER 22

CYLINDERS—CYLINDER HEADS

22-1. Cylinders.—*Materials* used for engine cylinders are close-grained gray cast iron, or nickle cast iron, or so-called *semi-steel,* ranging in tensile strength from 25,000 to 50,000 psi and having elastic limits in tension from 10,000 to 30,000 psi.[1]

In large engines cast-steel cylinders are used, but also with cast-iron liners usually. In airplane engines the liners are often turned of alloy-steel forgings to obtain very light cylinders.

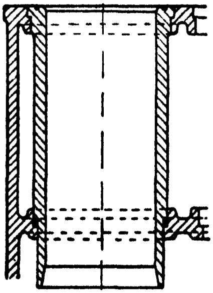

FIG. 22-1.—Wet liner.

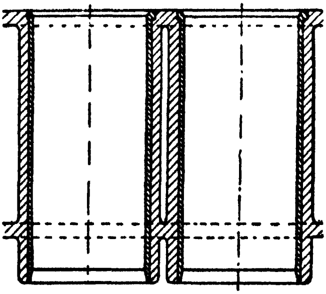

FIG. 22-2.—Dry liner.

Constructions.—Smaller engines have the cylinder and water jacket cast in one piece; in very small engines the cylinder, jacket, and frame may be combined in one piece. Sometimes, where the cylinder and jacket are cast in one piece, they are connected only at one end to allow for free expansion of the cylinder. All larger engines and most high-speed engines have separate liners. This construction takes care of the longitudinal expansion and also permits the use of a better grade, wear-resisting cast iron for the liner and a cheaper grade for the jacket.

Cylinder liners are made of two types, the *wet* type, when the outside is in direct contact with water, as in Fig. 22-1, and the *dry* type, pressed into the cylinder proper, Fig. 22-2. A dry liner is simpler to replace, and there is no danger of water leakage into either the combustion space or the crankcase. Its disadvantage is a decrease of the heat conduction through the composite wall. In some two-stroke compression-ignition oil engines, the liner is combined with the water jacket, Fig. 22-3, to make sure that the water will not leak into the cylinder through the joint.

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[1] MALEEV, V. L., "Machine Design," p. 51, International Textbook Company, Scranton, Pa., 1939.

Cylinders of multicylinder engines are cast either separately or in block, even in larger sizes.

Surface Treatment.—The wear of cylinder liners is substantially reduced if the inner surface is made harder. This can be accomplished by three different methods. One method is applied to dry liners and uses cast iron with 2.00 per cent nickel and 0.65 per cent chromium. After preliminary machining the liners are heated in an oven furnace to about 1550 F, oil quenched and for 2 hr tempered at 350 F. Finished they have a hardness of about Rockwell C 50. The wear of such liners is about one-third that of alloy cast-iron cylinders having a hardness of 230–240 Brinell.

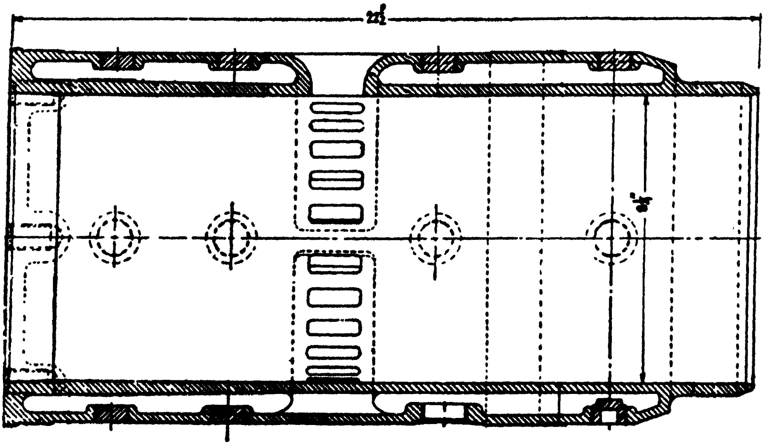

FIG. 22-3.—Liner with integral water jacket.

A second method is to heat cast-iron liners from inside by a high-frequency induction current and to quench them with water.[1] This gives a surface hardness of Rockwell C 52-55, which is subsequently slightly lowered by tempering. The depth of hardening is about 0.070 in., but the main body of the liner material is left unchanged. After tempering, the bore is honed to the final finish.

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[1] Induction Heat Treatment of Cylinder Liners, *Diesel Power*, June, 1941, p. 490.

A third method consists in chrome-plating electrically the inner bore, using a special procedure that gives a very hard but porous surface.[1] This surface helps the formation and retaining of a good oil film which results in an exceedingly low wear.

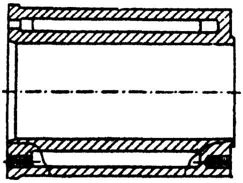

FIG. 22-4.—Cylinder of a small gas engine.

Hardening of the liner surface by either method reduces also the wear of the piston rings and piston surfaces.

Horizontal Cylinders.—A one-piece gas-engine cylinder of a small engine is shown in Fig. 22-4. A liner used with a combined water jacket and frame is shown in Fig. 22-3. A liner combined with part of the water jacket and part of the cylinder head is shown in Fig. 22-5. Its symmetrical design, effective water jacketing, and uniform distribution of material are responsible for comparatively low heat stresses and resultant good service.

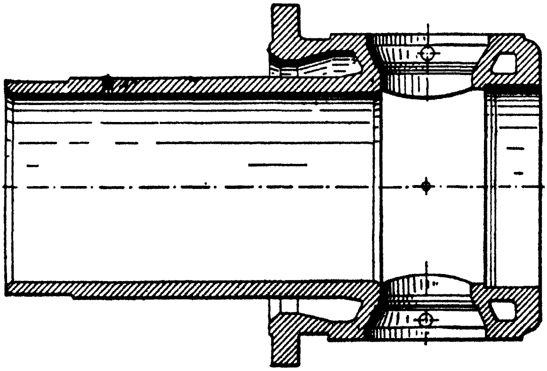

FIG. 22-5.—Gas-engine slip-in cylinder.

A cylinder of a double-acting engine is shown in Fig. 22-6. The thin, dry liner *d* of high-grade iron can be exchanged after it is worn at a fraction of the cost of a new cylinder, or of reboring the cylinder and fitting a new piston. The weak point of incorporating the valve receptacles into the cylinder ends is the tendency to develop cracks in the throat connections *f*; large fillets which give some flexibility and a separate two-piece water jacket decrease this tendency; *e* is an opening for the air starting valve; *g*, *g* for the spark plugs; *h*, *h* are oil grooves.

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[1] VAN DER HORST, Chrome Hardening of Cylinder Bores, *Mech. Eng.*, July, 1941, p. 536.

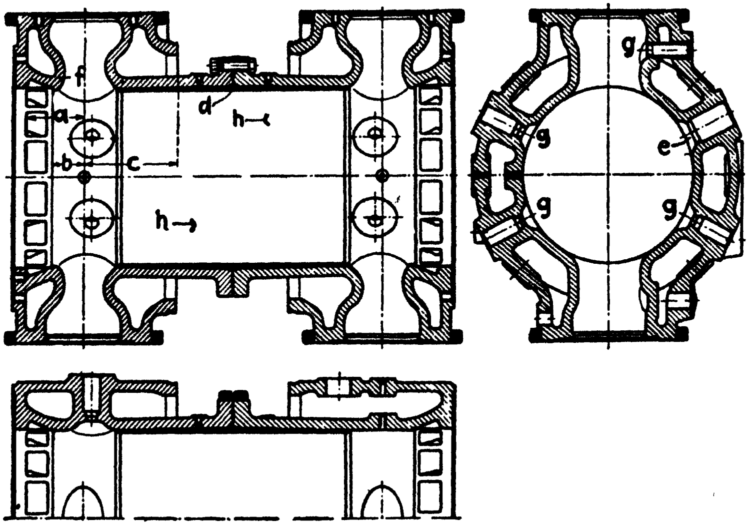

FIG. 22-6.—Cylinder of a double-acting four-stroke horizontal gas engine with dry liner.

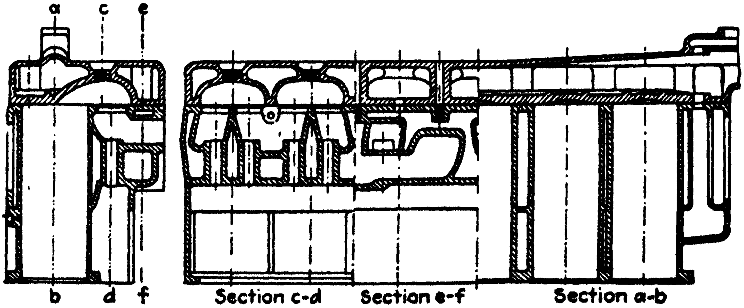

FIG. 22-7.—Cylinder block of a six-cylinder automobile engine.

Vertical Cylinders.—Figure 22-7 shows cast-in-block cylinders of an automobile engine with high-turbulence L-heads, and Fig. 22-8 shows a cylinder of an overhead-valve gasoline engine.

A typical cylinder with a liner for a gas or high-compression oil engine is shown in Fig. 22-9. A cylinder of a double-acting two-stroke oil engine is shown in Fig. 22-10.

Radial cylinders of airplane engines do not differ much from the general arrangement, except for the air-cooling fins, Fig. 22-11. The cylinder barrel is made of a steel forging, screwed and shrunk into the aluminum cylinder head.

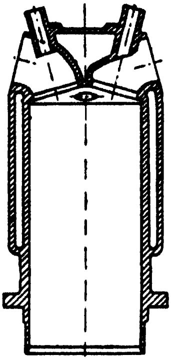

FIG. 22-8.—Valve-in-head cylinder.

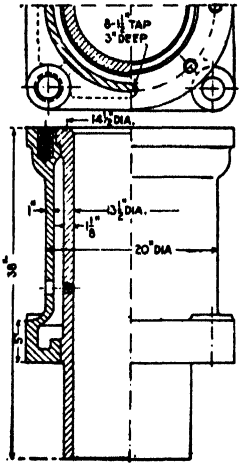

FIG. 22-9.—Vertical cylinder with liner.

Strength Computations.—The maximum stress in the wall of a cylinder liner can be computed by Birnie's formula for a thick-wall cylinder with open ends. In such a cylinder the longitudinal stress is zero and the maximum tangential or hoop stress is

$$s = p[(1 - \mu)d_i^2 + (1 + \mu)d_o^2]/(d_o^2 - d_i^2) \qquad (22\text{-}1)$$

where p is the maximum inner pressure, psi,
μ is Poisson's ratio,
d_i is the inside diameter, in.,
d_o is the outside diameter, in.

Solving equation (22-1) for the wall thickness $b = 0.5(d_o - d_i)$ gives

$$b = 0.5d_i\left[\sqrt{\frac{s + (1 - \mu)p}{s - (1 + \mu)p}} - 1\right] \qquad (22\text{-}2)$$

In determining the necessary wall thickness b the value of the design stress S_d is used for s in expression (22-2). In order to decrease the temperature stresses, the cylinder walls should be made as thin as possible, which calls for comparatively high stresses. Using a safety factor of 2 referred to the elastic limit, this gives for the materials indicated before a total stress $S_d = 5000$ to 15,000 psi after combining with the temperature stresses.

If the force acting on the piston is transmitted through the cylinder wall, as in a cylinder without a liner, Fig. 22-4 or Fig. 22-8, the hoop stress will be slightly lower; nevertheless expressions (22-1) and (22-2) can be used, as the results will be close enough.

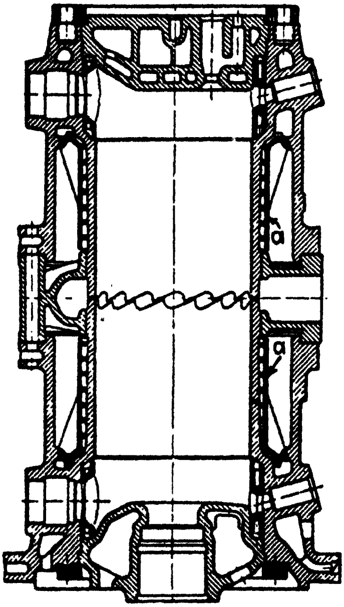

FIG. 22-10.—M. A. N. oil-engine cylinder, 33½ in. × 42 in., 160 rpm.

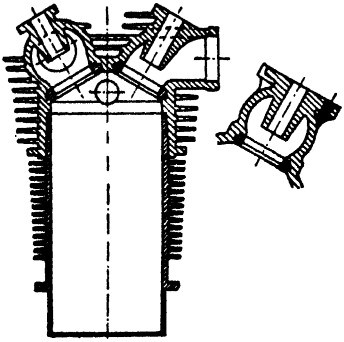

FIG. 22-11.—Cylinder of a radial engine.

The cylinder-wall thickness of automobile engines or other engines without liners, as in Fig. 22-4, the thickness b, expression (22-2), must be increased, adding about $\frac{1}{16}$ in. for a small-bore engine and up to $\frac{1}{8}$ in. for larger bore engines for future reboring.

In trunk-piston engines the inner cylinder wall, or the liner, is subjected to bending by the normal component of the piston pressure. However, this stress is very small and, occurring when the inside pressure begins to decrease, may be neglected.

For liners of oil engines the American Bureau of Shipping rules give

$$b \geqq D/15 \tag{22-3}$$

near the combustion space and through 20 per cent of the stroke.

Dry liners are supported on their full length, and their thickness can be made

$$b' = 0.030D \text{ to } 0.035D \tag{22-4}$$

Their outside diameter is made to give a medium-force fit, meaning an interference of about $0.0005D$.

The inner walls of automobile-engine cylinders usually are made with

$$b = 0.045\,D + \tfrac{1}{16} \text{ in} \tag{22-5}$$

The jacket-wall thickness is made

$$b_2 = 0.032D + \tfrac{1}{16} \text{ in.} \tag{22-6}$$

The water space c, or distance between the outside of the cylinder and inside of the jacket, can be set for engines of all types approximately as

$$c = 0.08D + \tfrac{1}{4} \text{ in.} \tag{22-7}$$

but must be checked for water velocity as indicated in Sec. 21-3.

The thickness of the flanges of the liners and of the water jackets must be computed taking into account the bending moments created by the hold-down bolts.

22-2. Temperature Stresses in Cylinders.—The simplest and sufficiently accurate method for evaluating the stresses set up by temperature differences in a cylinder liner is to use equations (21-20) and (21-21).

Figure 22-12 gives an example of the temperature variation and distribution in the liner wall actually measured in an oil engine and shows the range of temperature differences which must be considered in the design of liners.

If, instead of the difference of temperatures, the heat flow q is known, the stresses can be determined accurately enough from the equations (21-23) and (21-24). The heat flow q can be estimated from diagrams, Fig. 21-4 or 21-5, or from actual test data.

Example 22-1.—Determine the temperature stress in a 15-in. cast-iron cylinder wall 1 in. thick. The maximum heat flow is 30,000 Btu/sq ft-hr.

The coefficient $k = 320$, $m = (15 + 2 \times 1)/15 - 1 = 0.134$, $\mu = 0.27$, and, from expression (21-22),

$$t_i - t_o = qb/k = 30{,}000 \times 1/320 = 93.8 \text{ F}$$

By expression (21-20) the stress, taking from Table 21-1 $a = 0.065 \times 10^{-4}$ and $E = 13{,}000{,}000$, is

$$s_o = 0.065 \times 10^{-4} \times 13 \times 10^6 \times 93.8(1 - 0.134/3)/2(1 - 0.27) = 5{,}190 \text{ psi}$$

This figure gives the approximate tensile stress in the outer fibers. The compressive stress in the inner fibers, by expression (21-21), is

$$s_i = 0.065 \times 10^{-4} \times 13 \times 10^6 \times 93.8(1 + 0.134/3)/2(1 - 0.27) = 5{,}670 \text{ psi}$$

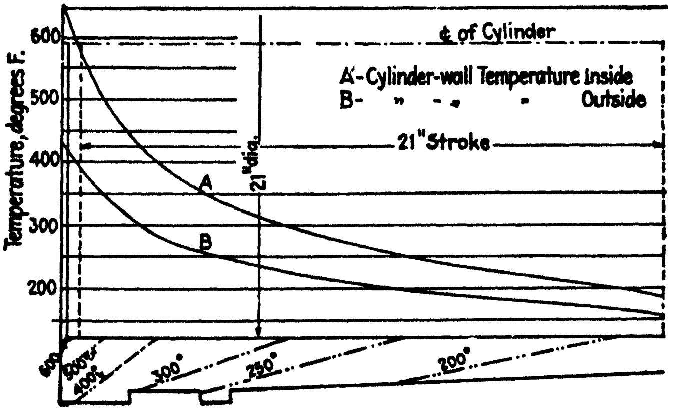

FIG. 22-12.—Temperatures in an oil-engine cylinder liner.

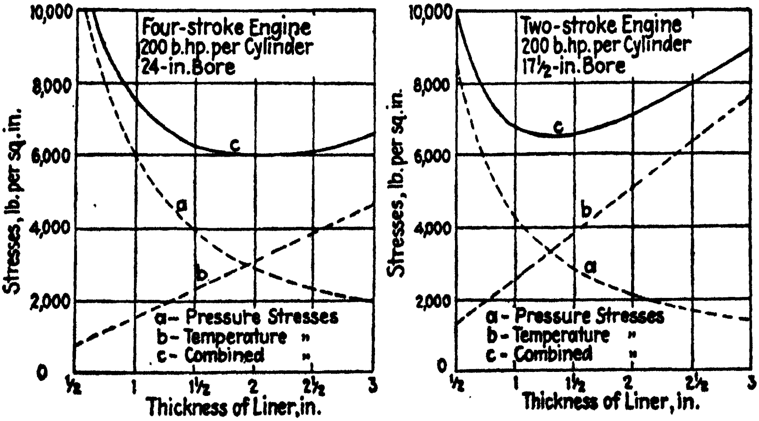

FIG. 22-13. FIG. 22-14.

FIGS. 22-13 and 22-14.—Stresses in cylinder liners.

Total Liner Stress.—Equations (21-20) to (21-24) show that the temperature stress is proportional to the temperature difference, to the wall thickness, and to the modulus of elasticity of the wall metal. On the other hand, the cylinder wall is subject to tensile stresses produced by the gas pressures. These latter stresses are also a function of the wall thickness *b*, decreasing with its increase. These pressure stresses relieve part of the compressive stresses in the inner fibers of the wall but increase the tensile stresses in its outer fibers. The total stress of the cylinder wall is the result of the combined temperature and pressure stresses.

Figure 22-13 illustrates the relation between these stresses for a four-stroke cylinder and Fig. 22-14 the same relation for a two-stroke cylinder of the same power. On account of the higher average temperatures in a two-stroke engine, the temperature stresses are considerably higher and result in higher combined stresses. These diagrams show clearly that the temperature stresses increase so rapidly with the size of cylinder and resulting wall thickness that they set a limit to the cylinder bore if the combined stress in cast iron is not to exceed 10,000 psi. They also show that the customary increase of the thickness of the cylinder liner at the end, where the highest temperatures prevail, is apt, by creating high-temperature stresses, to weaken the liner instead of strengthening it.

There exist two methods of decreasing temperature stresses. One is to make the cylinder wall thinner by using a material with a higher tensile strength than ordinary cast iron, such as semisteel or nickel cast iron, but with approximately the same modulus of elasticity The other method is to lower the temperature difference between the inside and outside wall surfaces. Experience has shown that in water-cooled cylinders an increase of the water velocity reduces failures in cylinder liners caused by temperature stresses. As a result special designs have been developed. In one, Fig. 22-10, the liner is surrounded by a shield *a* with spiral coils, which increase the water velocity and create turbulences favorable to heat transfer. In another design the spiral ribs are cast integral with the liner, thus not only creating greater water velocity but also presenting an additional cooling surface.

Expansion.—In order to prevent water leaks at the free end of a wet liner some sort of expansion joint must be used.

Figure 22-15 shows several expansion joints: *a* is simply a close fit of the liner to the bore in the jacket with no provisions to stop a leak if one should develop; *b* has grooves on the liner which are supposed to act as a labyrinth gland; *c* and *d* have rubber rings slightly compressed between flat or conical surfaces; and, finally, *e* is a regular stuffing box with a gland such as is used only in very large engines. Natural-rubber rings are satisfactory for temperatures up to 225 F; synthetic-rubber rings can be used up to 350 F.

Double-acting liners can be made to expand inward, Fig. 22-10. Cylinders of four-stroke horizontal engines may have a split water jacket, which allows the cylinder with the valve seats to expand freely, as shown in Fig. 22-6. The heat expansion of cylinders of large horizontal engines, particularly in tandem arrangement, is considerable. To prevent bending stresses and warping, the cylinders should be so fastened to the bedplate as to permit expansion by sliding on machined cast-iron surfaces.

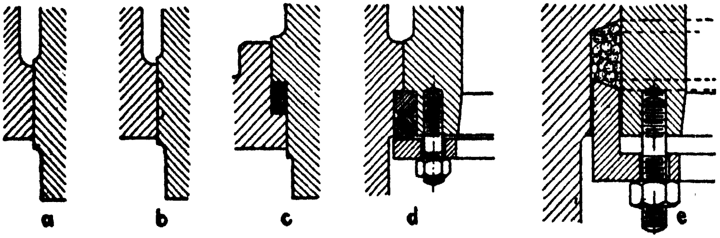

FIG. 22-15.—Types of liner expansion joints.

22-3. Cylinder Heads. *Points of Design.*—Due to frequent failure in service and difficulties experienced in manufacture, cylinder heads containing the inlet and exhaust valves are considered difficult pieces to design. The main causes of failure are:

1. Excessive rigidity, which does not permit expansion of the hot exhaust passage and leads to cracking of the bottom plate.
2. Small core holes, which give poor access to the interior for removing precipitated scale.
3. In oil engines, thin uncooled walls between the recesses for fuel and exhaust valves, which are apt to crack when the engine is overloaded.

These difficulties can be overcome by the following means:

1. Elimination of internal ribs, as the tubular walls which accommodate the valve cages provide sufficient support between the inside and outside plates.
2. Provision of handholes, for cleaning scale and venting for the core when casting,

3. Use of cast-iron, steel, or monel-metal tubes pressed in or cast into the recesses for the fuel valve (in oil engines) and hold-down studs in order to provide better water circulation.

4. Use of cast steel for the heads of larger engines or welding them of steel plates.

Materials.—The cylinder heads are usually made of close-grained cast iron or nickel cast iron. In order to avoid excessive rigidity a total tensile stress of 5000 psi may be allowed for gray iron and 8000 psi for nickel cast iron if the stress can be determined with a certain accuracy.

In heads of large engines made of low-carbon cast steel a stress of 12,000 psi may be allowed. The same stress may be used for simple cast-steel automobile covers.

Air-cooled cylinder heads with cooling fins are made of aluminum alloys on account of the high heat-transfer coefficient of aluminum. This gives a cooler combustion chamber and allows higher compression ratios without danger of detonation. The allowable stress may be from 5000 to 8000 psi depending on the alloy used.

Strength.—The accurate determination of the thickness of a head usually is difficult on account of its complicated shape, and several simplifying assumptions must be made.

If the cylinder head is a flat plate, its thickness h can be found from the expression

$$h = cD\sqrt{p/S_d} \tag{22-8}$$

where D is the cylinder bore, in.,
p is the maximum pressure, psi,
S_d is the allowable stress, psi,
c is a constant, in this case equal to 0.31.

A water-cooled cylinder head should be tested hydraulically under a pressure of 100 psi applied to the water space to make sure that the casting has no blowholes or porous places.

Temperature stresses in cylinder heads can be computed in a manner similar to that for liners. Figure 22-16 gives the distribution of temperatures as found by measurements in a cylinder head of a two-stroke oil engine.[1] These temperatures, together with measurements of heat carried away by the cooling water, make possible the determination of the surface heat-exchange coefficients which were found as $h_1 = 70$ Btu/sq ft-°F -hr for the inside and $h_2 = 307$ for the outside surface.

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[1] *Sulzer Tech. Review* (Winterthur, Switzerland), 1926, No. 2, pp. 1–11.

Example 22-2.—Find the temperature stress in a cast-iron cylinder head with temperatures as shown in Fig. 22-16.

Applying equation (21-19) and taking the temperature difference

$$t_i - t_o = 514° - 302° = 212 \text{ F}$$

from Fig. 22-16 gives

$$s = 0.5 \times 13{,}000{,}000 \times 0.0000065 \times 212/(1 - 0.27) = 12{,}270 \text{ psi}$$

Relieving Stresses.—This example shows how high the temperature stresses may run. Adding to this the pressure stresses which also run to about 2000 to 3000 psi reduces the factor of safety to a very low figure of about 1.25 in two-stroke engines and not over 1.75 in four-stroke engines in which the temperatures are slightly lower. This explains the frequent cracking of cylinder heads, particularly in larger gas and oil engines.

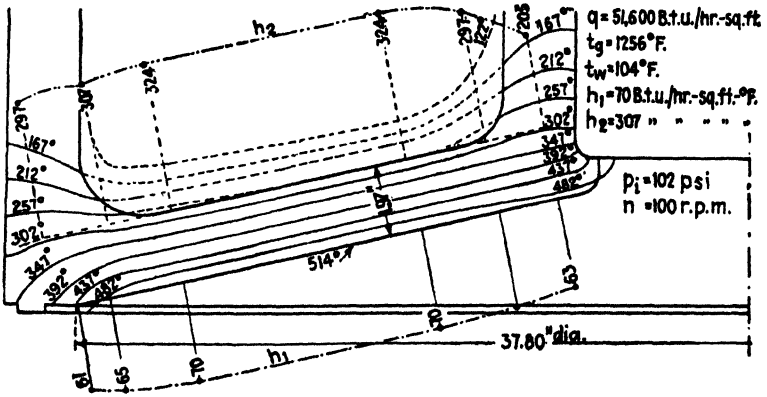


FIG. 22-16.—Temperatures and surface coefficients in a cylinder head at a two-stroke oil engine.

The best remedy is to use materials with higher elastic limits such as nickel or electric-furnace cast iron and to decrease the wall thickness. The use of cast steel for cylinder heads instead of cast iron reduces the thickness and hence the temperature difference to about one-half. However, the higher modulus of elasticity of steel again increases the stress, equation (21-19); and very little, if anything, is gained as far as the final safety factor is concerned.

Joints.—Three typical methods of securing a tight joint between the cylinder and its cover are shown in Fig. 22-17. Sketch *a* shows a joint with a gasket *g* as used in automobile and small gas engines. Sketch *b* shows a ground joint *h* used in large gas engines. Sketch *c* shows the joint with a recess *i* in the cylinder or liner and a corresponding ring projecting on the head; this type of joint is used in high-pressure oil engines.

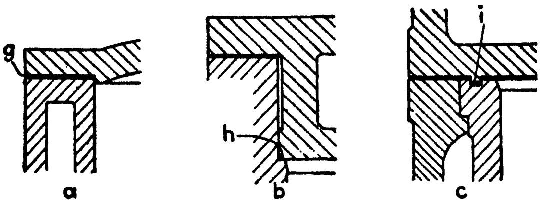


Fig. 22-17.—Types of cylinder-head joints.

It is well to remember that the diameter D_1 of the recess should be made as small as possible as it determines the force which is acting upon the hold-down studs, $F = 0.7854D_1^2p$. These studs should be computed allowing a stress of only about 6000 psi because of the high initial stress set up when tightening the nuts. The number of studs is selected approximately

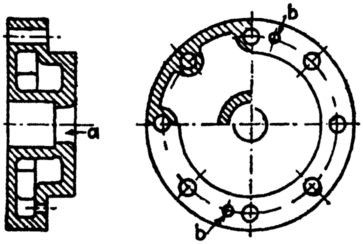


Fig. 22-18.—Cylinder head of a horizontal gas engine.

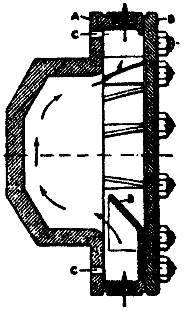


Fig. 22-19.—Gas-engine cylinder head.

$$i = 0.38D + 4 \tag{22-9}$$

Horizontal Engines.—A typical gas-engine head or cover is shown in Fig. 22-18: *a* is the place for the air starter valve; *b*, *b* are the cooling water connections.

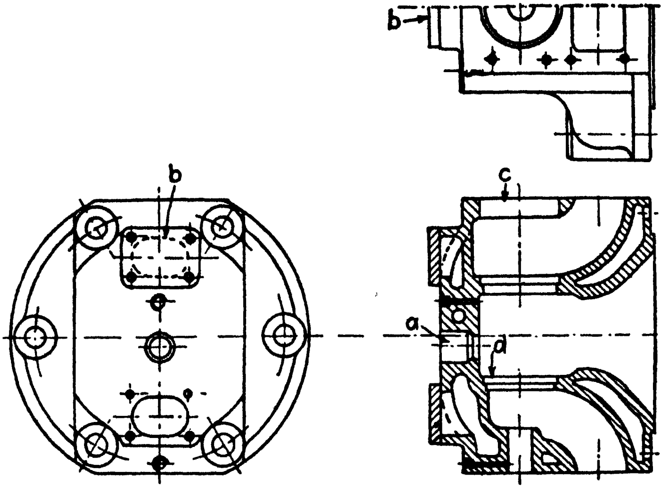


FIG. 22-20.—Cylinder head of a horizontal gas engine.

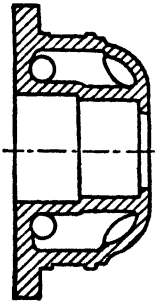
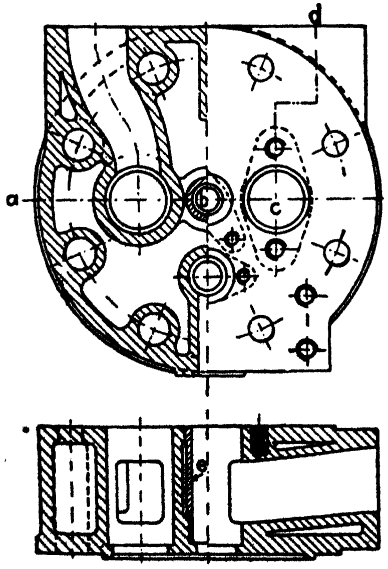


FIG. 22-21.—Cylinder head of a double-acting engine



Section a-b **Section c-d**

FIG. 22-22.—Cylinder head of a vertical oil engine.

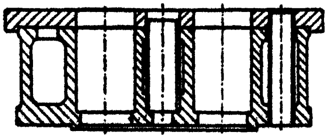


FIG. 22-23.—Cylinder head of a large oil engine.

Frequent cause of cracking cylinder heads through excessive temperature stresses is the depositing of scale from the cooling water. Scale acts as a heat insulator, decreases the heat transfer, and increases the mean wall temperature. Frequent inspection and elimination of scale increase the life of cylinder heads. Proper inspection and cleaning plates must be provided. Figure 22-19 shows the cylinder head of a medium-size gas engine. The removable plate *B* makes possible cleaning of the inside of the head. A deflector *D* directs the cooling water toward the hottest part of the cylinder. As in cylinder liners, effective cooling increases the life of cylinder heads.

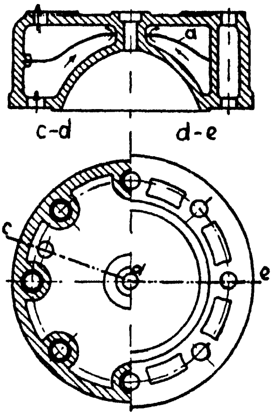


Fig. 22-24.—Cylinder head of a two-stroke compression-ignition oil engine.

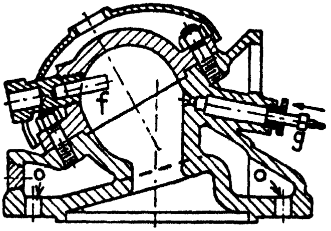


Fig. 22-25.—Cylinder head of a two-stroke hot-ball oil engine.

A cast-steel cylinder head of a larger single-acting gas engine is shown in Fig. 22-20, where *a* is the place for the air starter valve; *b*, *b* are clean-out holes; *c* is the place for the intake-valve cage and *d* for the exhaust-valve seat. Figure 22-21 shows the main section of a head for a double-acting engine whose cylinder was shown in Fig. 22-6.

Vertical Engines.—Figure 22-22 shows a typical head of a medium-size oil engine with a pressed-in monel-metal bushing *e* for the fuel valve. Figure 22-23 shows a head with a separate upper plate as used for larger oil engines and cast-iron bushings for the fuel nozzle and hold-down studs.

A typical cast-steel high-turbulence cover of a six-cylinder L-head automobile engine may be seen in Fig. 22-7. A cylinder head of a medium-size return-flow-scavenging two-stroke compression-ignition oil engine is shown on Fig. 22-24. An interesting feature is the cast-in pressed-steel shield *a* with an opening in the center to give a turbulent flow of cooling water and to cool especially well the fuel nozzle. Finally, Fig. 22-25 shows a cylinder head of a two-stroke hot-bulb oil engine made up of two main parts: the lower, water cooled, and the upper forming the hot bulb igniting the fuel injected by the nozzle *g*. When the engine is being started the fuel ignition is effected by the glowing igniter *f*, which is replaced at each starting from cold. The upper part of the hot bulb has a hood protecting it from excessive cooling; the lower part is water cooled through openings *o*, *o*, connecting it to the cylinder jacket.

22-4. Problems.—1. Determine the wall thickness of a 12-in. gray cast-iron gas-engine cylinder; the maximum pressure is 300 psi; take into account the temperature stresses.

2. Determine the wall thickness of a 9-in. gas-engine cylinder made of nickel cast iron, taking into account also the temperature stresses; the maximum gas pressure is 320 psi.

3. Determine the wall thickness of an 18-in. compression-ignition oil-engine wet cylinder liner. Select the material; take into account all stresses and compare with the results, using the rules of the American Bureau of Shipping.

4. Determine the wall thickness of a wet cast-iron cylinder liner with a $\frac{3}{16}$ in. thick, dry cast-iron liner inside for a $8\frac{1}{4}$-in. bore compression-ignition oil engine with a maximum cylinder pressure of 750 psi.

5. Determine the wall thickness of a $2\frac{7}{8}$-in. bore automobile-engine cylinder, Fig. 22-7.

6. Determine the wall thickness for an automotive-type cylinder, Fig. 22-8, with a bore of $3\frac{1}{8}$ in. Find the outside diameter of the uncooled cylinder part and of the water jacket and the latter's wall thickness.

7. Determine the temperature stress in a cast-iron cylinder liner $1\frac{1}{4}$ in. thick with a maximum heat flow of $q = 45{,}000$ Btu/sq ft-hr; also find the inside temperature of the liner if the outside temperature is 170 F.

8. Determine the thickness of a cylinder head, Fig. 22-18, for the engine of problem 1.

9. Determine the number and size of hold-down studs for the cylinder head for the engine of problem 3.

10. Determine the wall thickness of the cylinder head, Fig. 22-24, for a 13-in. × 15-in. compression-ignition oil engine. The maximum cylinder pressure is 725 psi.

11. Determine the number and size of studs for the cylinder head of problem 10.

CHAPTER 23

FRAMEWORK

23-1. Bedplates. *Horizontal Engines.*—A bedplate for a single-acting horizontal engine is shown schematically in Fig. 23-1. The force of explosion F puts the sides of the frame in tension and bending. The tensile stress is

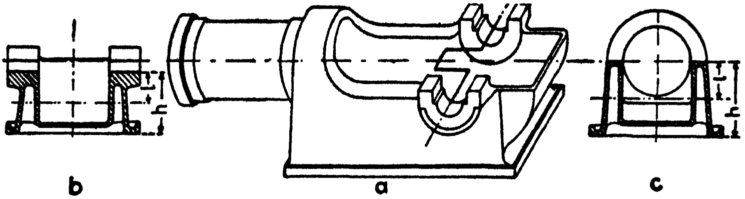


FIG. 23-1.—Bedplate of a horisontal single-acting engine.

$$s_1 = F/A \tag{23-1}$$

where A is the cross section, Figs. 23-1*b* and 23-1*c*, respectively. The bending moment is

$$M = Fl \tag{23-2}$$

where l is the distance from the center line of the cylinder to the center of gravity of the bedplate sides. Designating the moment of inertia of the frame section by I and the section modulus by $Z = I/c$ gives the bending stress

$$s_2 = M/Z \tag{23-3}$$

The total stress will be a maximum in the upper fibers of the section under the crankshaft,

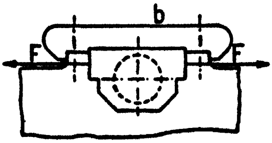


FIG. 23-2.—Steel bearing cap.

$$s = s_1 + s_2 \tag{23-4}$$

In order to reduce this stress, bearing caps may be made with lugs which are fitted accurately over the bedplate bosses, Fig.

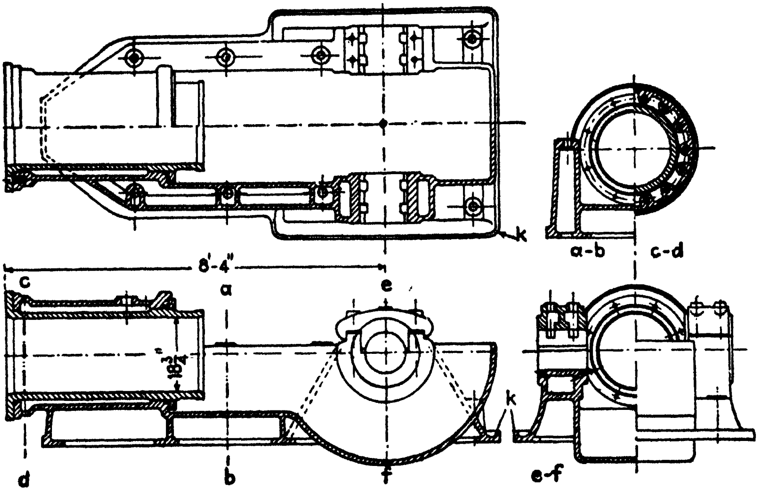


Fig. 23-3.—Bedplate of an 18¾-in. × 24-in., 220-rpm single-acting gas engine.

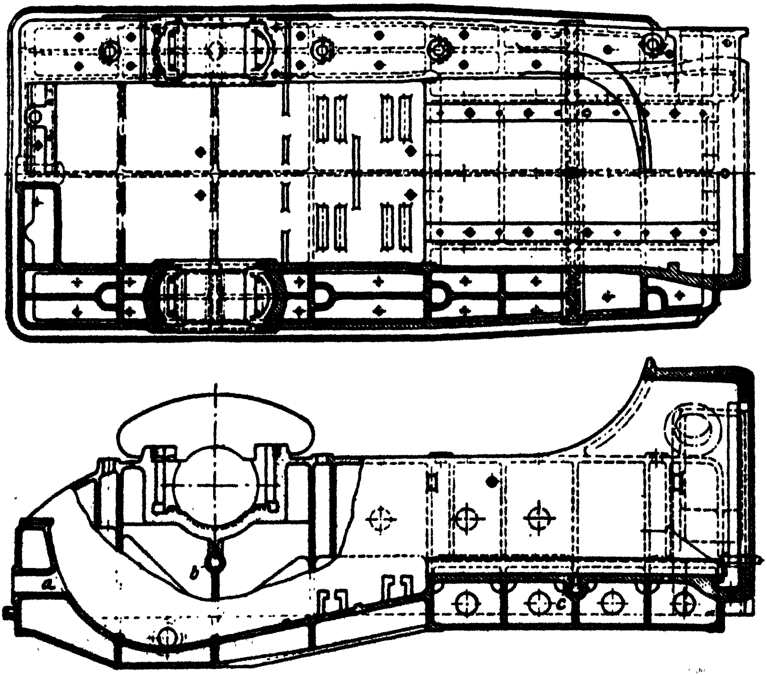


Fig. 23-4.—Bedplate of a double-acting gas engine. Scale 1:48.

23-3 or 23-4. Such caps should be cast of steel or made of a steel forging *b*, Fig. 23-2.

Expression (23-2) shows the advantage of raising the upper edge of the bed. Figure 23-3 shows a bedplate similar to Fig. 23-1 for a 125-hp single-cylinder gas engine, with an inserted cylinder liner, drawn to scale.

The turned-up lip *k* is necessary to catch the oil and thus to protect the concrete foundation.

Figure 23-4 shows the two main views of a bed for a double-acting gas engine with flat crosshead guides, cast integral with the bed.

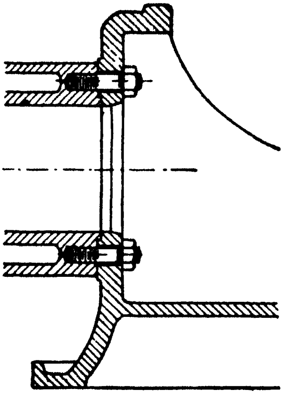


Fig. 23-5.—Fastening of a cylinder to the bed.

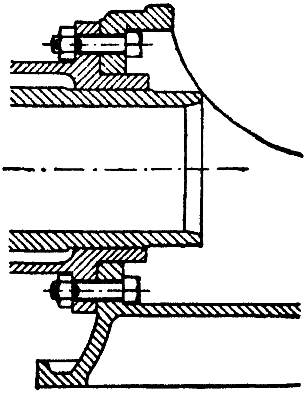


Fig. 23-6.—Fastening of a cylinder with liner to the bed.

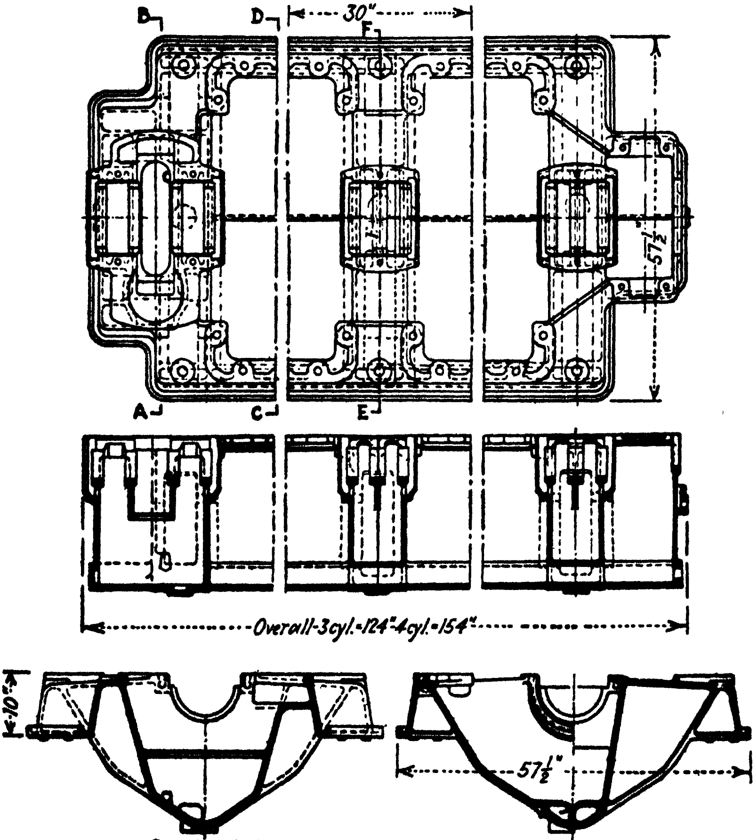
Figure 23-5 shows the method of fastening the cylinder to the bed, as used with both small single-acting and large double-acting engines. Figure 23-6 shows a method of fastening to the bed of a cylinder with a liner. A cylinder jacket cast integral with the bed, as used in medium-size engines, is shown in Fig. 23-3.

The diameter of the fastening bolts or studs is made from $0.09D$ for small down to $0.07D$ for large engines, where D is the engine bore. Their number is determined by assuming an allowable tensile stress of about 12,000 to 15,000 psi at the root of the thread. The initial stress from tightening the nuts should give a load about 25 per cent greater than $F_{\max}$.

An overhanging cylinder, Fig. 23-1, 23-5, or 23-6, if not supported at the other end by a special leg, stresses the upper fastening bolts more than the lower ones.

Vertical Engines.—The design of a bedplate for this type includes the following points:

1. Proper main bearings.
2. A girder construction under each main bearing for support
3. A strong and rigid connection between the bearing girders



Section A-B **Section C-D** **Section E-F**

Fig. 23-7.—Bedplate of a compression-ignition oil engine with an A-frame.

4. Provision for taking the pull of the working pressures.
5. An oiltight sump for collecting drainage of lubricating oil.
6. Machined surfaces for the frames or the crankcase and different auxiliary parts.

A typical bedplate for a vertical oil engine with an A-frame is shown in Fig. 23-7. The requirements of obtaining a good casting usually give a bedplate which is strong enough so far as the acting forces are concerned. The general thickness of metal is made about $h \geqq 0.04D + \frac{1}{4}$ in., where D is the cylinder bore, and this is increased by about $0.02D$ at the machined surfaces. An important problem in designing the bed is to make it rigid enough by proper ribbing.

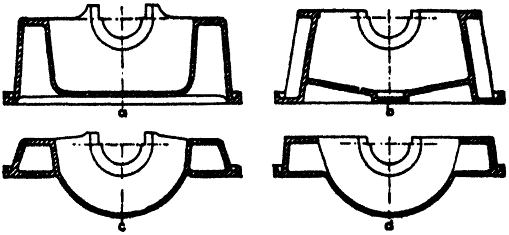


FIG. 23-8.—Types of bedplates for vertical engines.

Figure 23-8 gives approximate proportions for different types of cast-iron bedplates. Figure 23-8*a* is a section of a bedplate for an A-frame stationary engine; Fig. 23-8*b* is a section of a bedplate for a stationary engine with a continuous enclosed crankcase; Figs. 23-8*c* and 23-8*d* are the corresponding sections for main marine engines in which it is desirable to lower the center of gravity of the engine.

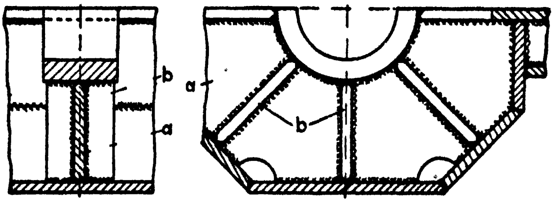


FIG. 23-9.—Welded bedplate and main bearing.

In lightweight welded steel bedplates the question of rigidity is of particular importance. While the support *a*, Fig. 23-9, may be sufficient for strength, ribs *b*, *b* are very essential for rigidity.

Regardless of the amount of rigidity incorporated in the construction of the bedplate, a long bedplate, as of a six- or eight-cylinder oil engine, will warp or sag if not supported at a sufficient number of points while being installed.

Automobile engines are usually built without a separate bedplate, the latter being combined with the crankcase; the crossties containing the main bearings are inverted and the bearing caps hold the crankshaft. Oil pans of sheet metal or a light aluminum casting close the crankcase from the bottom, Figs. 23-25 and 23-26, and also act as lubricating-oil coolers.

23-2. Main Bearings.—The design of the main bearings involves two main problems: to reduce the wear to a minimum and to provide means of taking up the wear after it occurs. The wear is kept low as long as a lubricating-oil film separates the journal from the bearing. A film of oil is able to sustain a heavy pressure for a short time, but when the film has broken down, if the engine has been standing still for a certain length of time, a comparatively small pressure is sufficient to cause abrasion. The surfaces cannot receive a new film of oil until the pressure is removed or until a sufficient peripheral speed is developed. Therefore, the lubrication of single-acting four-stroke-engine main bearings present considerably less difficulty than that of two-stroke engines, because the pressure on the journals of a four-stroke engine is continuously reversed. When the peripheral speed of the journal is so high as to reduce appreciably the oil viscosity through the heat generated by friction, pressure lubrication must be used.

In a single-acting two-stroke engine the direction of pressure in most cases is not reversed, and proper lubrication is therefore more difficult. As a rule the maximum bearing pressure should be kept about one-third lower than that in a similar four-stroke engine. In the case of high peripheral speeds or high pressures, pressure lubrication becomes necessary and in extreme cases water cooling of the bearings also.

Oil Grooves.—A correct arrangement of oil grooves is very important in order to obtain proper bearing lubrication. Incorrectly located grooves usually are worse than none. Axial and spiral grooves, Fig. 23-10, may be used for lengthwise oil distribution. However, a better practice for lengthwise oil distribution is to use only side relieves, where the two shells meet, Fig. 23-10 or 23-14. Such a side relief not only serves as an oil reservoir and distributor but helps to wedge the oil in between the surfaces and thus to maintain the necessary oil film. Neither lengthwise nor circumferential, or spiral, grooves should extend to the pressure region of the bearing surface. To avoid scraping off of the oil the edges of oil grooves must be well rounded.

Horizontal Engines.—In medium-size low-speed engines, round shells with horizontal, Fig. 23-11*a*, or inclined, Fig. 23-11*b*, caps are used. The wear can be taken up by means of shims.

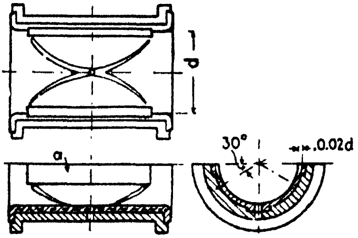


FIG. 23-10.—Main bearing shell.

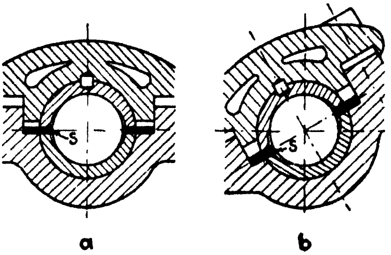


FIG. 23-11.—Arrangement of bearing shells and caps.

Larger, especially double-acting, engines are equipped with four-piece bearings with take-up in the horizontal direction by means of wedges, Fig. 23-12*a*, or screws, Fig. 23-12*b*.

Ring lubrication for main bearings, used rather extensively some years ago, Fig. 23-7, is now used only for small- and medium-size horizontal engines and for crankcase-scavenging two-stroke engines, which do not lend themselves as well to pressure lubrication. An oil-ring outboard bearing for a low-speed engine is shown in Fig. 23-13.

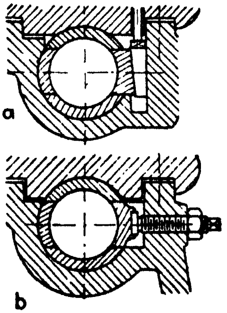


FIG. 23-12.—Bearing adjustment.

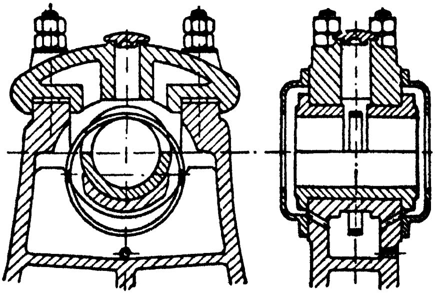


FIG. 23-13.—Oil-ring bearing.

Vertical Engines.—Most vertical engines, regardless of sizes, are equipped with pressure-feed lubrication. The oil is admitted either from underneath, Fig. 23-14, or through the cap, Fig. 23-15. Theoretically the latter construction is better, as the oil is delivered at the point where the pressure is at a minimum; the disadvantage is in the more complicated oil piping and in the necessity of breaking the pipe connection every time the bearing shell must be taken out.

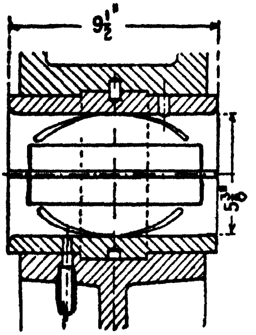


FIG. 23-14.—Pressure lubricated bearing.

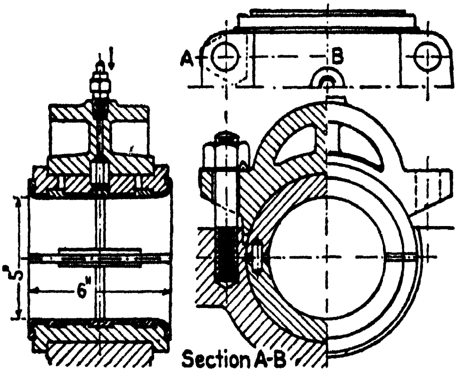


FIG. 23-15.—Pressure lubricated bearing of an oil engine.

Figure 23-16 shows the so-called *precision main-bearing shells* of a heavy-duty high-speed compression-ignition oil engine. These shells are precision machined and are installed without any hand scraping of the inner surface; however, in larger engines some fitting to the bearing is necessary. They have no take-up, and when a shell is worn it is replaced by a new one.

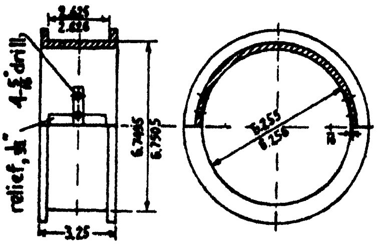


FIG. 23-16.—Precision main-bearing shells for a heavy-duty oil engine.

In single-acting low-speed vertical engines the wear occurs more at the bottom of the lower shell; therefore, main bearings of these engines have only two shells with a take-up by shims between the shells.

Bearing Shells.—In low- and medium-speed engines, the shells are made of brass or bronze in small engines, of cast iron in medium-size, and of steel in large engines, all lined with babbitt. The chief essentials are:

1. Proper thickness of the backing.
2. Good quality of babbitt—a sufficient hardness to stand pressure, and toughness to prevent cracking of the lining.
3. Good bond between the babbitt and the backing material. This is obtained by tinning the shells before babbitting and sometimes putting in swallowtail keys if a heavy babbitt layer is used.

The thickness of the backing is made about 0.20 d for cast-iron shells, 0.12 d for cast steel, and 0.10 d for bronze. The thickness of the babbitt layer should be about 0.03 d for small engines. decreasing to 0.02 d for large engines, where d is the journal diameter. In automotive and aviation engines the trend is toward very thin babbitt, about $\frac{1}{32}$ in. and sometimes only 0.005 in. thick.

Round shells should be ground from the outside to exact dimensions in order to ensure perfect contact with the bed. A good fit helps considerably the scraping in of the shells, prevents cracking of babbitt, and provides proper transfer of the friction heat. In high-speed engines the shells, Fig. 23-16, are made of a thin steel or bronze backing, about 0.04 d to 0.05 d, with a very thin white-metal or copper-lead lining.

Ball and Roller Bearings.—The use of ball and roller bearings in internal-combustion engines is becoming more and more popular. Their advantage lies in:

1. The low coefficient of friction.
2. The absence of wear under favorable conditions.
3. The independence in lubrication.

The main cause hampering their still wider use is the difficulty of their application to multithrow crankshafts. Some single-cylinder gas engines are built with taper roller bearings as main bearings, and practically all radial airplane engines are built with ball bearings.

A very important requirement for the satisfactory operation of a ball or roller bearing is a perfect rigidity of the crankshaft. Even a small flexure is apt to cause a disagreeable growling noise and result in a failure of the bearings.

23-3. Frames and Crankcases. *Vertical Engines.*—A large number of different types of frames are used, but only the main types will be mentioned here.

The A-type frame, so called from its transverse section, Fig. 23-17, is the earliest type of diesel-engine frame and is still sometimes used with low-speed medium-size oil and gas engines.

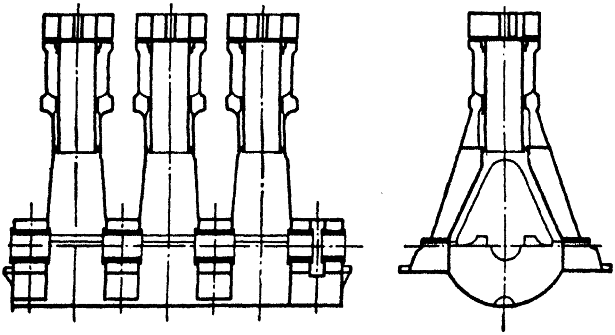


FIG. 23-17.—A-type of oil-engine frame.

Each cylinder stands on its own legs, unsupported by the other cylinders. The legs of the column are cast in one piece with the cylinder jacket into which the liner is fitted. This construction reduces machining and fitting operations to a minimum.

A typical B- (or box-) frame, often called a *crankcase*, is shown in Fig. 23-18 and is used for all higher speed engines. Sometimes

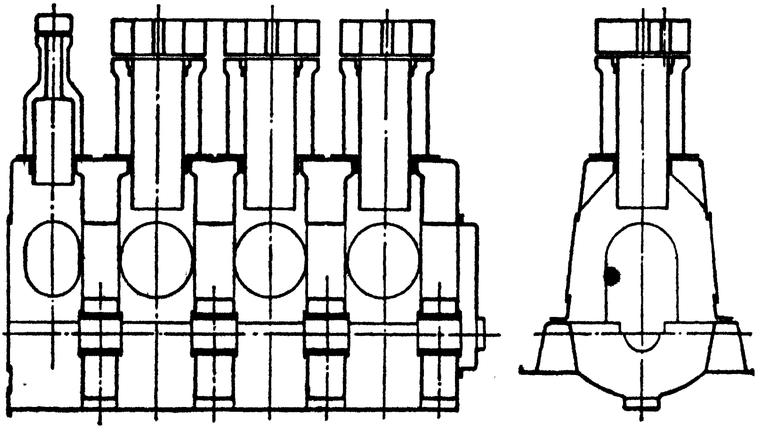


FIG. 23-18.—B-type frame of an air-injection oil engine.

the box or girder construction of the crankcase is utilized for transmitting the stresses from the cylinder to the bedplate. A better construction is to provide steel rods for this purpose,

Fig. 23-19. The crankcase is, in this case, practically a spacer between the cylinder and bedplate. However, the casting should be properly ribbed as the stresses from tightening the hold-down rods may cause serious distortion. If the cylinders are cast separately, they are fastened either by a round flange with studs or by passing the hold-down rods through each corner of a deep square flange of hollow section cast as a part of the cylinder jacket. In large engines the crankcase is divided into two or more pieces, sometimes a separate piece under each cylinder, the separate castings being bolted together and forming a continuous box of great strength and rigidity.

In order to reduce the engine weight, welded crankcases are used. Figures 23-20 to 23-22 illustrate the building of a welded one-piece bedplate and crankcase for a twelve-cylinder V-type lightweight 1000-hp oil engine.[1]

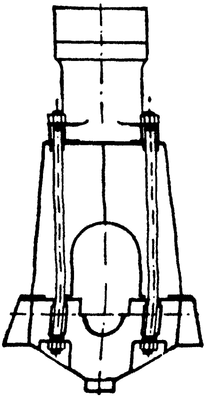


Fig. 23-19.—Crankcase with tie bolts.

A different approach is used in the engine frames built by the Steel-Barrel Co., Ltd., of Usbridge, England, from the designs of C. H. Stevens; the frames are used mainly for marine engines. The gas-pressure loads are taken by plates *a*, Fig. 23-23, one between each pair of cylinders and at each end. Two long plates *b* are flame cut so as to be inserted through the rectangular openings of plates *a*, as shown in the upper center of Fig. 23-23, and are welded to the plates *a*. The top and bottom seats for

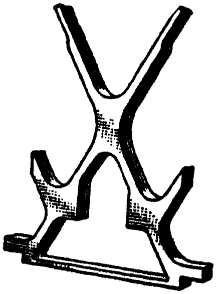


Fig. 23-20.—Flame-cut part of a crankcase.

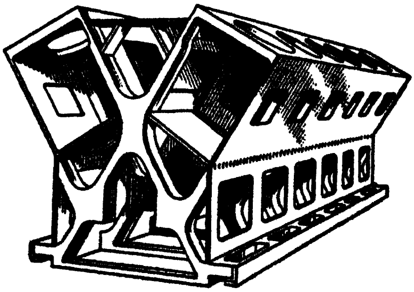


Fig. 23-21.—Crankcase in process of fabricating by welding.

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[1] Chapman, E., *Motorship*, September, 1933.

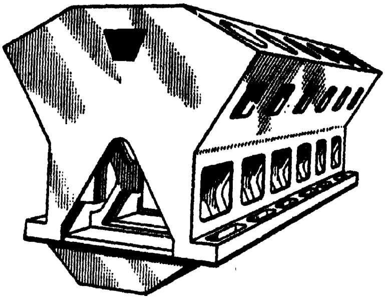


FIG. 23-22.—Finished welded crankcase.

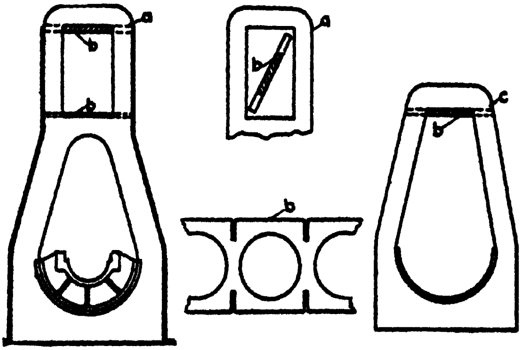


FIG. 23-23.—Parts of a welded engine frame, designed by C. H. Stevens.

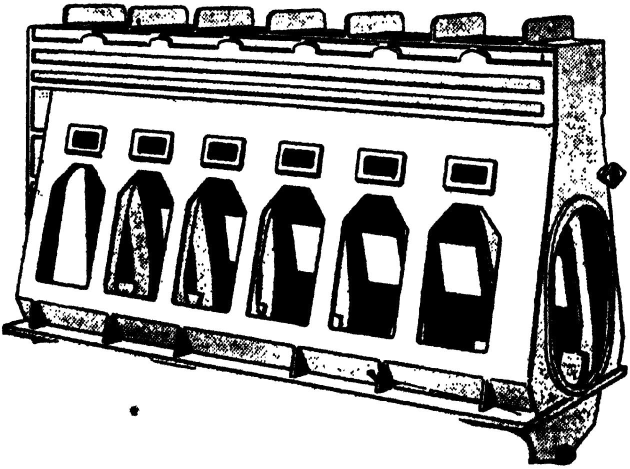


FIG. 23-24.—Assembled welded engine frame, designed by C. H. Stevens.

the liners are machined in plates *b*. The bearings are welded to the plates *b*. An assembled frame of a six-in-line engine is shown in Fig. 23-24. If individual cylinders with water jackets are used, the design is modified: only one lower plate *b* is used, the shape of the vertical plates is changed to *c*, Fig. 23-23, and the cylinders are bolted to the plate *b*.

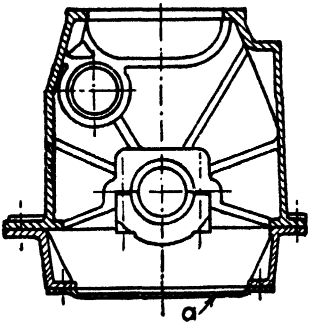


FIG. 23-25.—Automobile-engine crankcase

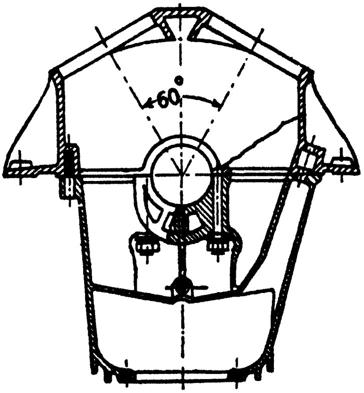


FIG. 23-26.—Crankcase of a V-type engine.

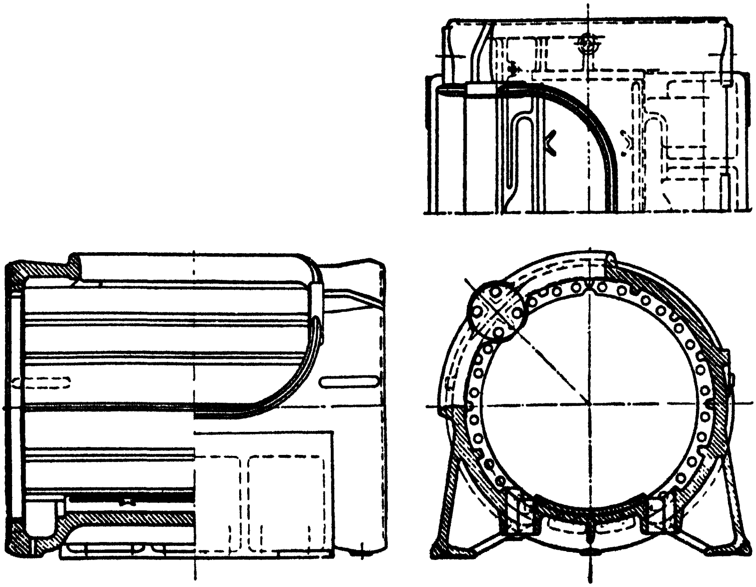


FIG. 23-27.—Intermediate crosshead guide of a tandem engine.

A welded-frame construction saves from 30 to 45 per cent in the weight of an engine.

Figure 23-25 shows a section of the crankcase of an automobile engine with the main bearings incorporated into the crankcase, without a bedplate and with only an oil pan *a*.

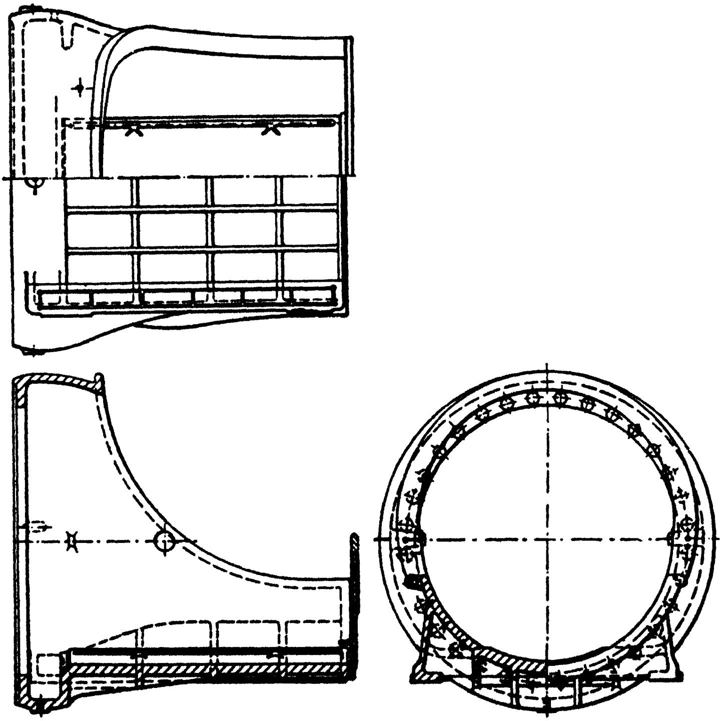


Fig. 23-28.—Crosshead guide for protruding piston rod of a horizontal double-acting gas engine.

Figure 23-26 shows a cross section of an aluminum crankcase for a V-type automotive or aircraft engine.

Horizontal Engines.—What can be classified as a frame of a horizontal engine, in addition to the bedplate, is the intermediate crosshead guide of a tandem engine, Fig. 23-27, which at the same time is a tie piece, and the guide for the crosshead supporting the end of the piston rod extended through the back cover, Fig 23-28.

23-4. Problems.—1. Determine the number and size of the studs necessary to fasten the cylinder of a 14-in. gas engine to the bedplate, Fig. 23-5;

the explosion pressure is 320 psi. Assume: (*a*) that the cylinder is overhanging and (*b*) that it is supported at the outer end.

2. Determine the number and size of bolts necessary to fasten the cylinder of a 16-in. compression-ignition oil engine to its bedplate, Fig. 23-5; the firing pressure is 700 psi. Assume: (*a*) that the cylinder is overhanging and (*b*) that it is supported at the outer end.

3. Find the thicknesses at the rough and machined surfaces of the bedplate walls of a vertical 17-in. bore engine: (*a*) when cast of cast iron class 25 and (*b*) welded of steel plates SAE 1020, assuming the same strength as when made of cast iron; (*c*) compare the bedplates (*a*) and (*b*) as to rigidity.

4. Work problem 3 for a 12-in. bore engine.

5. Determine the thickness of the babbitt lining and the outside diameter of the cast-iron shell, Fig. 23-10, for a 7-in. shaft.

6. Determine the thickness of the babbitt lining and the outside diameter of the bronze-babbitted bearing shells, Fig. 23-16, for a 4-in. crankshaft.

7. Determine the thickness of the white-metal lining and the outside diameter of the steel bearing shells, Fig. 23-16, for a 5-in. crankshaft.

8. Determine the size of the tie bolts, Fig. 23-19, to hold the cylinder shown on Fig. 22-9; assume a firing pressure of 700 psi.

9. Determine the number and size of the studs to fasten the cylinder, Fig. 22-8, of a 4¼-in. × 5-in. oil engine to its crankcase, Fig. 23-25; assume a maximum pressure of 725 psi.

CHAPTER 24

RECIPROCATING-ROTARY MOTION

24-1. Piston Motion. *Piston Travel.*—The piston position, or its travel x from a dead center, is according to Fig. 24-1

$$x = r(1 - \cos \alpha) \pm l(1 - \cos \beta) \tag{24-1}$$

where r is the crank radius, in., and
l is the length of connecting rod, in.

$\cos \beta$ can be eliminated, noticing that $l \sin \beta = r \sin \alpha$, or

$$\sin \beta = r \sin \alpha / l \tag{24-2}$$

and therefore

$$\cos \beta = \sqrt{1 - (r \sin \alpha / l)^2} \tag{24-3}$$

Applying the formula for series and considering only the first term gives, with an error of less than 0.1 of 1 per cent,

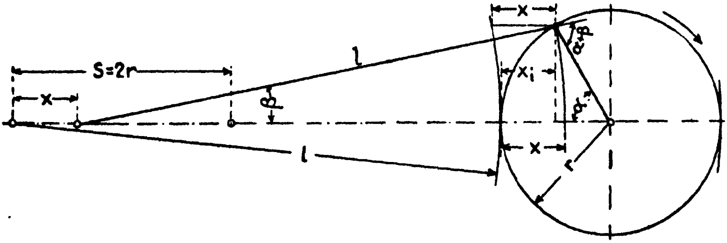


FIG. 24-1.—Piston and crank motion diagram.

$$\cos \beta = 1 - r^2 \sin^2 \alpha / 2l^2 \tag{24-4}$$

and expression (1-15) becomes

$$x = r[1 - \cos \alpha \pm \tfrac{1}{2}(r/l) \sin^2 \alpha] \tag{24-5}$$

The minus sign applies to the return stroke if the angle α is measured from the other dead center. Table 24-1 gives the piston positions in per cent of the stroke for the more common ratios of connecting-rod length to crank radius l/r.

Angularity of Connecting Rod.—For an infinite connecting rod, $r/l = 0$ and equation (24-5) becomes $x_i = r(1 - \cos\alpha)$. The difference between x and x_i is the so-called *correction for angularity,* $r^2 \sin^2 \alpha/2l$. It has the maximum value of $r^2/2l$ for $\alpha = 90°$.

TABLE 24-1.—PISTON POSITIONS IN PERCENTAGE OF STROKE

Degrees from top center	Ratio of connecting-rod length to crank length l/r					
α	3.75	4.00	4.25	4.50	4.75	5.00
10	1.0	1.0	0.9	0.9	0.9	0.9
20	3.8	3.7	3.7	3.7	3.6	3.6
30	8.4	8.3	8.2	8.1	8.0	8.0
40	14.5	14.3	14.1	14.0	13.9	13.8
50	21.8	21.5	21.3	21.1	21.0	20.8
60	30.0	29.7	29.4	29.2	29.0	28.8
70	38.8	38.5	38.1	37.9	37.6	37.4
80	47.8	47.4	47.0	46.7	46.5	46.2
90	56.7	56.3	56.0	55.6	55.3	55.1
100	65.2	64.8	64.4	64.1	63.9	63.6
110	73.0	72.7	72.4	72.1	71.8	71.5
120	80.0	79.7	79.4	79.2	79.0	78.8
130	86.1	85.8	85.6	85.4	85.2	85.1
140	91.1	90.9	90.7	90.6	90.5	90.4
150	95.0	94.9	94.8	94.7	94.6	94.6
160	97.8	97.7	97.7	97.6	97.6	97.6
170	99.4	99.4	99.4	99.4	99.4	99.4

Velocities.—The piston velocity c is the first derivative of travel x with respect to time t, $c = dx/dt$, or by expression (24-5)

$$c = r[\sin\alpha \pm \tfrac{1}{2}(r/l)\sin 2\alpha]\, d\alpha/dt$$

Designating crank velocity by v, and noticing that $v\,dt = r\,d\alpha$,

$$d\alpha/dt = v/r \tag{24-6}$$

and

$$c = v[\sin\alpha \pm \tfrac{1}{2}(r/l)\sin 2\alpha] \tag{24-7}$$

Expressing the crank velocity in feet per second

$$v = 2\pi rn/(12 \times 60) = \pi rn/360 \tag{24-8}$$

where n is revolutions per minute, gives for the piston velocity

$$c = (\pi rn/360)\,[\sin\alpha \pm \tfrac{1}{2}(r/l)\sin 2\alpha] \tag{24-9}$$

From equation (24-9) the piston velocity corresponding to any crank position may be determined; c attains its highest value when $\alpha + \beta = 90°$; $c_{max} = \pi rn/360$; the average piston speed

$$c_a = 4rn/(12 \times 60) = rn/180$$

therefore

$$c_{max} = (\pi/2)c_a = 1.57c_a \tag{24-10}$$

24-2. Inertia Loads.—*Acceleration* a of the piston is obtained by differentiating the piston speed c with respect to time t, and with expression (24-7),

$$a = dc/dt = v[\cos \alpha \pm \tfrac{1}{2}(r/l) \cos 2\alpha]\ (d\alpha/dt)$$

and since, by expressions (24-6), changing v to fps,

$$d\alpha/dt = 12v/r$$

the acceleration in fpsps

$$a = 12v^2[\cos \alpha \pm (r/l) \cos 2\alpha]/r \tag{24-11}$$

Inertia.—The reciprocating masses must be accelerated from a zero velocity to the velocity v of the crankpin in the short period of approximately one-fourth of one revolution. The necessary force is obtained from the piston or the flywheel. This force F can be presented as a product of mass by acceleration, $F = ma$. If the total weight of the reciprocating parts is W lb, the weight per square inch of piston area $w = W/A$, and the mass per square inch of piston area $m = W/Ag$, then, by expression (24-11),

$$F = 12Wv^2[\cos \alpha \pm (r/l) \cos 2\alpha]/Agr \tag{24-12}$$

or substituting for v its expression (24-8)

$$F = 0.0000284rn^2W[\cos \alpha \pm (r/l) \cos 2\alpha]/A \tag{24-13}$$

Expression (24-13) shows the influence of the number n of revolutions per minute. As can be seen from the method of deducing this expression and expression (24-4), expression (24-13) is only approximate, the first term $0.0000284rn^2W \cos \alpha/A$ representing the first harmonics, the second term $0.0000284r^2n^2W \cos 2\alpha/lA$ representing the second harmonics, all higher harmonics being omitted.

TABLE 24-2.—VALUES OF INERTIA FACTOR [$\cos \alpha + (r/l) \cos 2\alpha$]

Degrees from top center α	Ratio of connecting-rod length to crank length, l/r					
	3.75	4.0	4.25	4.50	4.75	5.0
0	1.267	1.250	1.235	1.222	1.210	1.200
10	1.236	1.219	1.206	1.194	1.183	1.173
20	1.144	1.131	1.120	1.110	1.101	1.093
30	0.999	0.991	0.984	0.977	0.971	0.966
40	0.812	0.809	0.807	0.804	0.803	0.801
50	0.597	0.599	0.602	0.604	0.606	0.608
60	0.367	0.375	0.382	0.389	0.395	0.400
70	0.138	0.150	0.162	0.172	0.181	0.189
80	−0.077	−0.061	−0.047	−0.035	−0.024	−0.014
90	−0.267	−0.250	−0.235	−0.222	−0.210	−0.200
100	−0.425	−0.408	−0.395	−0.382	−0.372	−0.361
110	−0.546	−0.543	−0.522	−0.512	−0.503	−0.495
120	−0.633	−0.625	−0.618	−0.611	−0.605	−0.600
130	−0.689	−0.686	−0.684	−0.681	−0.679	−0.677
140	−0.723	−0.723	−0.725	−0.727	−0.729	−0.731
150	−0.733	−0.741	−0.748	−0.755	−0.761	−0.766
160	−0.733	−0.748	−0.760	−0.769	−0.779	−0.786
170	−0.733	−0.750	−0.764	−0.776	−0.787	−0.797
180	−0.733	−0.750	−0.765	−0.778	−0.790	−0.800

The forces of inertia F attain a maximum with $\alpha = 0$, or at the head-end dead center, when

$$F = 12Wv^2(1 + r/l)/Agr \tag{24-14}$$

The first term, $12Wv^2/Agr$, is the centrifugal force of the reciprocating masses when considered concentrated at the crankpin.

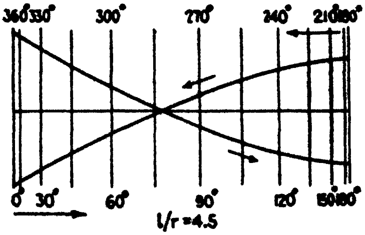


FIG. 24-2.—Acceleration or inertia curves.

Computing from expression (24-13) the values of F for consecutive angles α, the inertia curves can be plotted against crank angles. It is sufficient to take increments of 15°. To facilitate the computation, the factors [$\cos \alpha + (r/l) \cos 2\alpha$] are given in Table 24-2 for the commonly used ratios of l/r. The values of the factors for the return stroke, $\alpha = 180$ to $360°$, are the same except that they have opposite signs, Fig. 24-2. The abscissas in Fig. 24-2 may be computed by expression (24-5) or Table 24-1 or they can be found graphically.

Reciprocating Parts.—The reciprocating parts whose inertia must be considered in a single-acting engine consist of the piston with rings and wristpin and half of the connecting rod. In crosshead engines the weights of the piston rod and crosshead are added, and in double-acting engines to this must be added the tail rod and tail crosshead, and the weight of water in the piston, rods, and water connections.

The weights of these parts may be calculated from sketches for each part as it is designed. For preliminary computations it may be desirable to estimate the weights of the reciprocating parts, using Table 24-3 for this purpose.

TABLE 24-3.—WEIGHTS OF RECIPROCATING PARTS

Type of engine	Weight W/A, psi of piston area	
	Spark ignition (gas and gasoline)	Compression ignition (oil)
Single-acting engines:		
Aircraft	0.22– 0.40	0.30– 0.45
Automobile	0.30– 0.50	
Truck and tractor	0.35– 0.75	0.35– 0.75
Medium-speed stationary	1.5– 4	1.5– 4.5
Low-speed, short stroke ($l < 1.5d$)	4– 7	5– 8
Low-speed, long stroke ($l > 1.5d$)	6–10	8–12
With crosshead ($l < 1.5d$)	9–14	10–15
Double-acting engines:		
Single cylinder per crank	12–20	14–22
Tandem	20–26	

24-3. Forces. *Net Efforts.*—The action of forces of inertia upon the connecting rod is of the same nature as the action of the forces resulting from the gas pressures upon the piston. The simplest way of taking them into account is to combine them on the indicator card. This is the reason that the inertia load in expression (24-12) was referred to 1 sq in. of piston area.

In order not to obscure the indicator card, it is advisable to spread the consecutive strokes, Fig. 24-3, plotting positive pressures above and negative pressures below the atmospheric line.

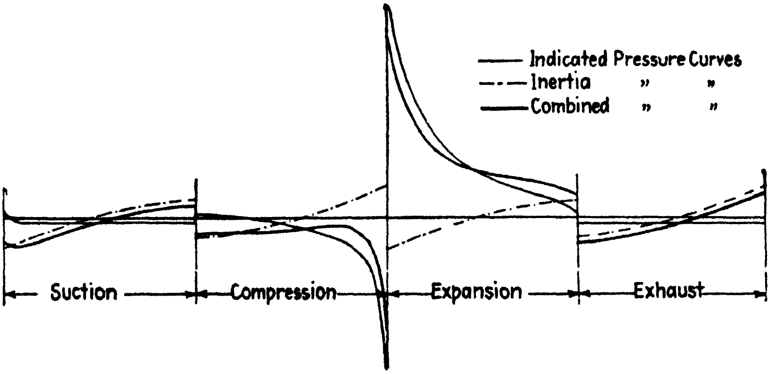


FIG. 24-3.—Net-effort diagram for a four-stroke, single-acting, single-cylinder gas engine.

Next are plotted the inertia loads in the same manner; and, finally, the curve of the combined pressures is drawn, the heavy lines in Fig. 24-3, which can be called *net effort*, transferred to the connecting rod.

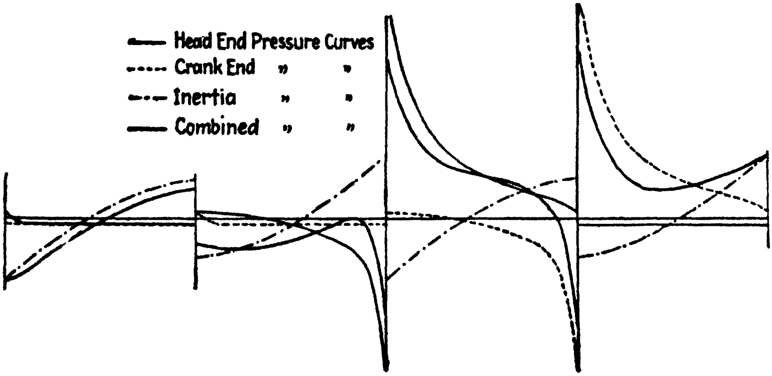


FIG. 24-4.—Net-effort diagram for a four-stroke, double-acting, single-cylinder horizontal gas engine.

In a double-acting engine the pressures acting on both sides of the piston must be plotted, and their differences represent the net gas pressures. These, combined with the inertia loads, give the net-effort curve, Fig. 24-4.

Figure 24-5 illustrates the procedure: 1-2 is the explosion and expansion line for the head end of the cylinder; 3-4 is the compression line for the crank end; and 5-6 the inertia curve of the reciprocating parts. The points of the net-effort curve 7-8 were obtained by adding the pressures and inertia loads as they appear with their signs. For any point e, its ordinate $ei = gi - ki + hi$.

In a tandem double-acting four-stroke engine, four pressure lines must be combined for each stroke—expansion, exhaust, suction, and compression, and the resulting net pressures are combined with the inertia forces to find the net efforts.

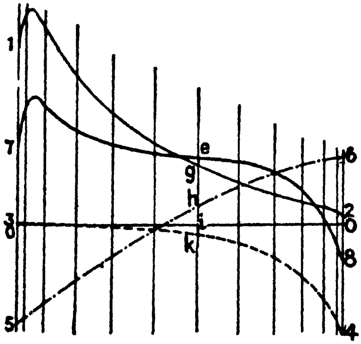


FIG. 24-5.—Combining pressures in a double-acting engine.

Forces.—The net effort F_p acting in the direction of the center line of the piston, when transferred to the connecting rod, is resolved into force F_c, Fig. 24-6, acting along the connecting rod and the normal force F_n, which acts upon the crosshead or trunk piston. At the crankpin the force F_c is again resolved into a tangential force F_t and a radial force F_r.

These forces can be expressed from Fig. 24-6 as follows:

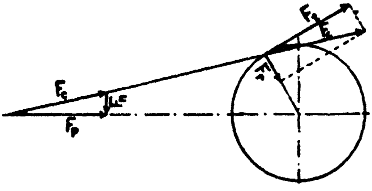


FIG. 24-6.—Diagram of acting forces.

The force along the connecting rod, using expression (24-4)

$$F_c = F_p/\cos\beta = F_p/(1 - r^2 \sin^2 \alpha/2l^2) \qquad (24\text{-}15)$$

The normal force, using expressions (24-2) and (24-4)

$$F_n = F_p \tan\beta = F_p r \sin\alpha/l(1 - r^2 \sin^2 \alpha/2l^2) \qquad (24\text{-}16)$$

The radial force

$$F_r = F_c \cos(\alpha + \beta) = F_p \cos(\alpha + \beta)/\cos\beta \qquad (24\text{-}17)$$

The tangential force, often called the *rotative effort,*

$$F_t = F_c \sin(\alpha + \beta) = F_p \sin(\alpha + \beta)/\cos\beta \qquad (24\text{-}18)$$

Noticing that $\sin(\alpha + \beta) = \sin\alpha\cos\beta + \sin\beta\cos\alpha$ and substituting for $\sin\beta$ and $\cos\beta$ expressions (24-2) and (24-4) give

$$F_t = F_p \sin\alpha\left(1 + \cos\alpha/\sqrt{(l/r)^2 - \sin^2\alpha}\right) \qquad (24\text{-}19)$$

24-4. Torque Diagram.—This is a short name for a diagram obtained by plotting tangential forces or efforts against crank travel.

The tangential efforts F_t can be computed from expression (24-19). For convenience values $\sin\alpha[1+\cos\alpha/\sqrt{(l/r)^2-\sin^2\alpha}]$

TABLE 24-4.—VALUES OF FACTOR $\sin\alpha[1 + \cos\alpha/\sqrt{(l/r)^2 - \sin^2\alpha}]$

Degrees from top center α	Ratio of connecting-rod length to crank length, l/r					
	3.75	4.00	4.25	4.50	4.75	5.00
0	0.000	0.000	0.000	0.000	0.000	0.000
10	0.219	0.216	0.214	0.212	0.210	0.208
20	0.428	0.423	0.418	0.413	0.410	0.407
30	0.616	0.609	0.602	0.597	0.592	0.587
40	0.777	0.768	0.760	0.753	0.747	0.742
50	0.900	0.891	0.884	0.877	0.871	0.866
60	0.985	0.977	0.970	0.964	0.959	0.954
70	1.028	1.022	1.017	1.013	1.009	1.005
80	1.032	1.029	1.026	1.024	1.022	1.020
90	1.000	1.000	1.000	1.000	1.000	1.000
100	0.938	0.941	0.943	0.946	0.948	0.950
110	0.851	0.857	0.862	0.867	0.871	0.874
120	0.747	0.755	0.762	0.768	0.773	0.778
130	0.632	0.641	0.648	0.655	0.661	0.666
140	0.509	0.518	0.526	0.532	0.538	0.543
150	0.384	0.391	0.397	0.403	0.408	0.413
160	0.256	0.261	0.266	0.270	0.274	0.278
170	0.128	0.131	0.134	0.136	0.138	0.139
180	0.000	0.000	0.000	0.000	0.000	0.000

are given in Table 24-4 for every 10° of crank travel and commonly used ratios of l/r. The values from this table must be multiplied by the value of F_p taken from the net-effort diagram for the respective crank angle.

The tangential forces F_t can be found by a simple, graphical method, Fig. 24-7: F_t is the segment of the vertical line drawn through point m until it intersects the extension of the connecting rod or the connecting rod itself, point o. To prove this, it is sufficient to notice that the angle $\overline{nlm} = \alpha + \beta$; distance $\overline{mn} = \overline{lm} \sin (\alpha + \beta) = F_p \sin (\alpha + \beta)$ and $\overline{om} = T = \overline{nm}/\cos \beta$, which gives $F_t = F_p \sin (\alpha + \beta)/\cos \beta$, or expression (24-18).

A tangential effort diagram using Fig. 24-3 for net efforts F_p is shown in Fig. 24-8. The base of this diagram is the distance traveled by the crankpin during the time the piston travels through one complete cycle. For a four-stroke engine the piston, as represented by the indicator card, travels 4*s* in. where *s* designates the piston stroke and the base of the torque diagram represents two revolutions or $2\pi s$. This diagram represents the force developed at the crankpin, and F_{tm} is the mean tangential force. If the same pressure scale is used in the indicator and the torque diagrams, the areas of both should be equal. This is a good check on the accuracy of the work.

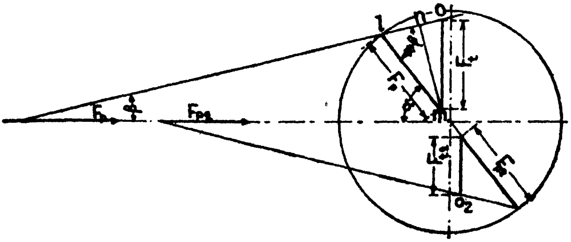


FIG. 24-7.—Determination of tangential forces.

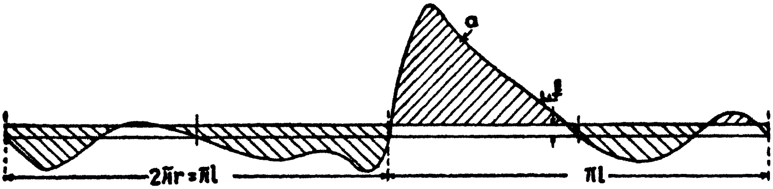


FIG. 24-8.—Torque diagram for net-effort diagram of Fig. 24-3.

Torque diagrams are used to determine the weight of the flywheel.

24-6. Problems.—1. Plot piston velocities against piston travel for $l/r = 3.75$, $l/r = 5$, and $l/r = \infty$; use increments of 30°; compare the curves for these l/r ratios; engine speed $n = 1200$ rpm, stroke is 6 in.

2. Plot piston velocities against piston travel for $l/r = 4$, $l/r = 4.75$, and $l/r = \infty$; use increments of 30°; compare the curves obtained. $r = 3$ in., $n = 1200$ rpm.

3. Determine the values of the unbalanced second harmonics at the upper and lower dead centers in a 4¼-in. × 5-in. two-cylinder vertical engine with cranks at 180°, at 1200 rpm; the weight of the reciprocating parts in each cylinder is 9 lb; the connecting-rod-crank ratio is 4:1.

4. Determine the value of the unbalanced second harmonics at the upper and lower dead centers for a marine four-cylinder four-stroke gasoline engine; its rated power is 75 bhp at 750 rpm. The ratio of the connecting-rod length to the crank radius is 4:1; the weight of the reciprocating parts is the same as in a truck engine.

5. Determine the values of the total forces of inertia at the upper and lower dead centers for the engine of problem 3.

6. Construct the inertia curves for the cylinder of a truck engine, Table 24-3, having a $4\frac{1}{2}$-in. bore, $5\frac{1}{2}$-in. stroke, and $11\frac{5}{8}$-in. connecting rod; assume an idling speed of 400 rpm, a normal speed of 1200 rpm, and maximum speed of 1500 rpm.

7. Construct the inertia curves for the cylinder of an automobile engine, Table 24-3, having a $2\frac{7}{8}$-in. bore, a $3\frac{1}{2}$-in. stroke and a $7\frac{7}{16}$-in. connecting rod; assume an idling speed of 500 rpm, a normal speed of 2800 rpm, and a maximum speed of 3500 rpm.

8. Plot a curve of normal pressures during the expansion stroke for the engine diagram used in Fig. 24-3, with $l/r = 5$; the pressure scale is 320 psi per inch.

9. Plot a torque diagram for the net-effort diagram given in Fig. 24-4; the pressure scale is 320 psi per inch; use 10° crank-travel increments.

CHAPTER 25

VIBRATION AND BALANCING

25-1. Vibration. *Causes of Vibration.*—If a force displaces a body from its position of rest and in so doing establishes a restoring force, the body will come back to its original position but, owing to its mass, will pass it, creating a restoring force in the opposite direction, and so on until these movements back and forth through the equilibrium position are damped out by friction. Such movements are called *vibration.*

If a body is set into vibration by forces recurring in regular intervals and these intervals are the same as the natural time period of vibration of the body or a simple fraction or multiple of this time period, the amplitude of vibration of the body may increase even if the forces are not large. Such conditions are termed *resonance* and may become dangerous for the operation or even existence of an engine.

There are three kinds of forces which may cause vibrations in an internal-combustion engine—horizontal and vertical forces due to the inertia of reciprocating parts, centrifugal forces due to unbalanced rotating parts, and tangential forces acting on the crankshaft pins.

Reciprocating Parts. *Harmonics.*—Using expression (24-12) and designating all constant factors by M, the force of inertia of reciprocating parts can be presented as

$$F = M[\cos \alpha \pm (r/l) \cos 2\alpha] \qquad (25\text{-}1)$$

Graphically this expression was represented by Fig. 24-2. If the abscissas in this diagram are changed from piston travel to crank travel, curve a, Fig. 25-1, is obtained. The two terms of expression (25-1) plotted separately give the curves b and c, both of which are sinusoidal curves. Curve b is called the *first harmonic* and represents the influence of the centrifugal force of the reciprocating masses considered concentrated at the crankpin, curve c is the *second harmonic* and represents the influence of the angularity of the connecting rod. Its frequency is twice the frequency of the first harmonic, but the amplitude is considerably shorter. If the centrifugal force is expressed not by an approximate formula but by an exact one, it can be resolved into a series, each term representing a sinusoidal curve with an increasing frequency and decreasing amplitude, such as curve *d*, which represents the next, or fourth, harmonic. The influence of still higher harmonics usually is negligible.

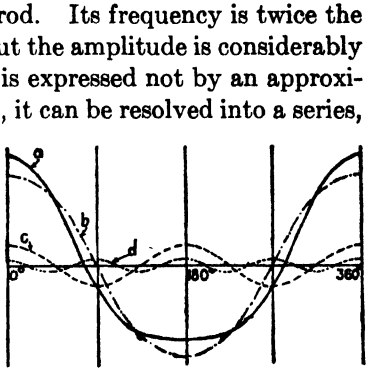


FIG. 25-1.—Diagram of inertia forces.

25-2. Method of Analysis.—A simple and comprehensive method of analyzing the harmonic motions of the piston and other reciprocating parts connected to it is obtained by substituting for their weight two imaginary weights. These weights, each one equal to one-half of the weight of the reciprocating parts, rotate in opposite directions with equal velocities, and in such a relation that they come together when the piston is at a certain dead center. This means that the center of gravity of the two imaginary weights always lies on the diameter of the crank circle coinciding with the center line of the cylinder, Fig. 25-2. As can be seen from Fig. 25-1, the pair of weights representing the first harmonic rotate at crankshaft speed, those representing the second harmonic, at twice the speed, the fourth at four times the speed, the sixth at six times the speed, and so on. However, the sixth and all higher harmonics are so small that they are seldom considered.

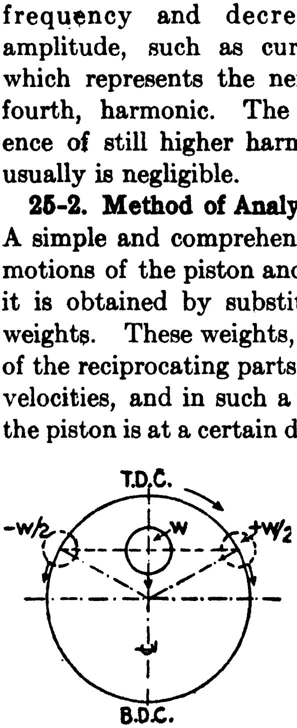
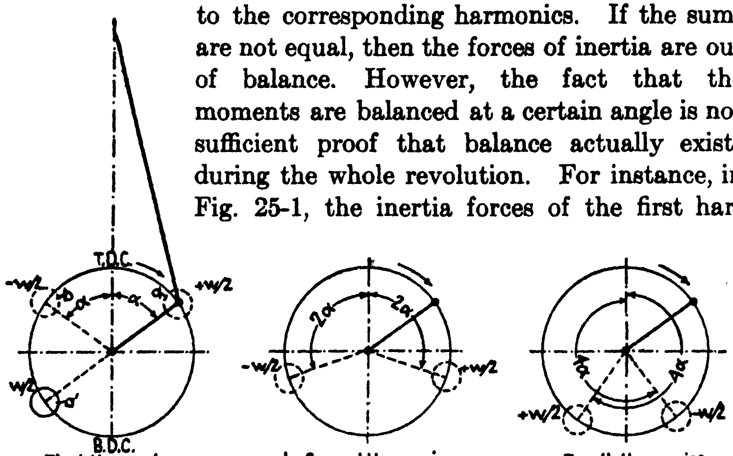


FIG. 25-2.—Representation of a simple harmonic.

The following simple rule will help to avoid confusion when using this method: For any given position of the cranks the positions of the imaginary weights are found first by determining the angles of travel of each crank from a certain dead center from the corresponding piston and second by moving their imaginary weights, one clockwise and the other counterclockwise by angles equal to the crank angle times the number of the harmonic—two for the second, four for the fourth, etc. All these angles must be counted from the same dead center. It is convenient to designate the weights rotating with the crank by a plus sign and those rotating in the opposite direction by a minus sign.

If the moments of both imaginary weights that rotate clockwise and anticlockwise about the line normal to the piston travel are such that the sum of the moments below the line is equal to the sum of the moments above it at any crank angle, then equilibrium exists and the forces of inertia are balanced in respect to the corresponding harmonics. If the sums are not equal, then the forces of inertia are out of balance. However, the fact that the moments are balanced at a certain angle is not sufficient proof that balance actually exists during the whole revolution. For instance, in Fig. 25-1, the inertia forces of the first harmonics may be zero at an angle of 90°, intersection of the line of abscissas by the curve *b*; or the forces of inertia of the second harmonics may be zero at 45 and 135°, intersection of the line of abscissas by the curve *c*, but the system as such is not balanced.



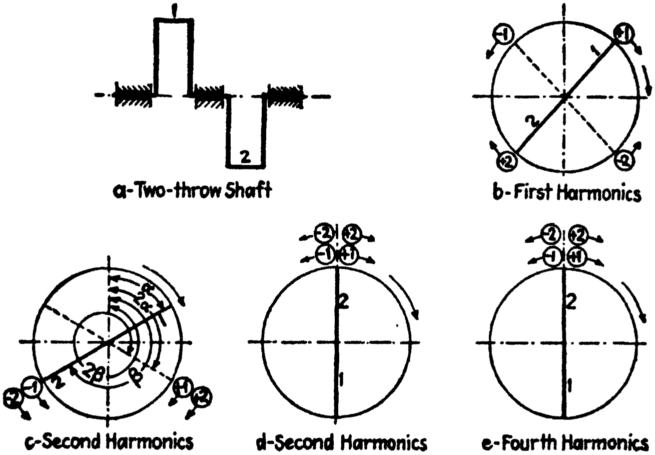
a-First Harmonics b-Second Harmonics c-Fourth Harmonics

Fig. 25-3.—Balance analysis of a single-cylinder engine.

This method is illustrated by the analysis of a single-cylinder and a two-cylinder engine.

Single-cylinder Engine.—The first harmonics of the inertia forces for this engine type are represented by weights *a* and *b*, Fig. 25-3*a*, clearly showing an unbalance. The imaginary weight *a* can be balanced by an equal weight at *a′* rotating with the crankshaft, but the weight *b* rotating in the opposite direction cannot be balanced by either an addition to or a subtraction of a weight from the crankshaft. When half of the weight of the reciprocating parts is counterbalanced, as by a weight *a′*, the unbalanced part of the first harmonics, due to the weight *b*, causes the engine to vibrate equally in all directions in the plane of rotation. In single-cylinder motor cycle engines a fore-and-aft unbalanced force is less objectionable than an up-and-down one. For this reason it is customary to overbalance the engine, *i.e.*, to use a counterweight whose effect is greater than one half the effect of the reciprocating parts. The second harmonics, Fig. 25-3*b*, and the fourth harmonics, Fig. 25-3*c*, show that the engine is unbalanced at these harmonics too.

The second and fourth harmonics of the inertia forces cannot be balanced by weights rotating at crankshaft speed.



a-Two-throw Shaft

b-First Harmonics

c-Second Harmonics

d-Second Harmonics

e-Fourth Harmonics

FIG. 25-4.—Balance analysis of a two-cylinder engine.

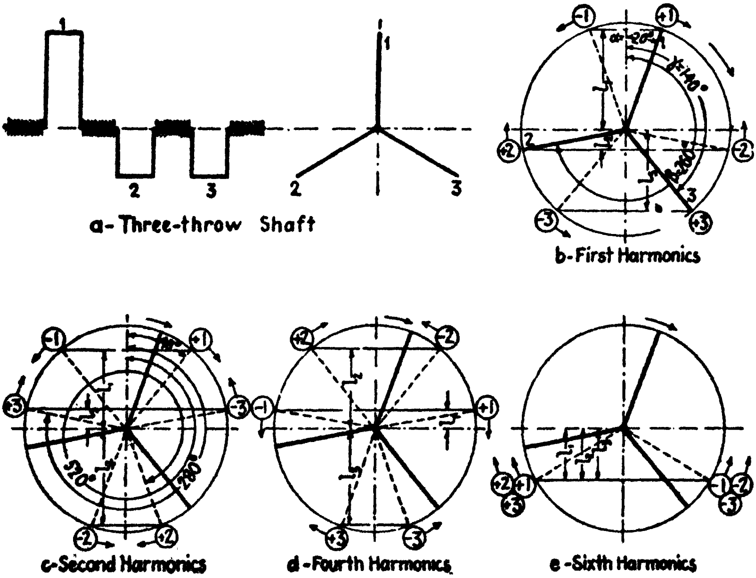
Two-cylinder Engine.—The first harmonics of the inertia forces of an engine with cranks at 180°, Fig. 25-4*a*, are represented in Fig. 25-4*b*, which shows that in any position the weight 1 rotating clockwise is balanced by weight 2 rotating in the same direction and so are, respectively, weights 1 and 2 rotating counterclockwise. Thus the first harmonics are inherently balanced, leaving only unbalanced couples set up by the forces on cranks 1 and 2. These couples can be balanced by attaching counterweights, as will be shown later.

Figure 25-4*c* shows the location of the imaginary weights of the second harmonic for about the same crank position as in Fig. 25-4*b*. The diagram was obtained by following the above given rule and shows that the weights are unbalanced. The unbalance becomes a maximum at the dead centers, Fig. 25-4*d*, which shows the top dead center. The unbalanced second harmonics cause a vibration in the plane of the center line of the cylinders with a frequency twice that of the engine speed.

The fourth harmonics, Fig. 25-4*e*, act the same as the second harmonics but have less force and a higher frequency.

25-3. Multicylinder Engines.—For a multicylinder engine it is simpler to check the balance analytically.

Example 25-1.—Check the balance of the forces of inertia of the reciprocating parts for a three-cylinder three-crank engine with the cylinder center lines in one plane, Fig. 25-5*a*.



a-Three-throw Shaft

b-First Harmonics

c-Second Harmonics

d-Fourth Harmonics

e-Sixth Harmonics

FIG. 25-5.—Balance analysis of a three-cylinder engine.

The cranks are 120° apart. Assuming for crank 1 $\alpha = 20°$ gives for crank 3 $\gamma = 20 + 120 = 140°$ and for crank 2 $\beta = 140 + 120 = 260°$, Fig. 25-5*b*.

Since the imaginary weights rotating clockwise are always located symmetrically to those rotating counterclockwise, and since all weights are equal, one can simply find, instead of an equilibrium of moments, the sum of the three distances l_1, l_2, and l_3 of the weights from the line drawn through the center of the crankshaft and normal to the cylinder center line, assuming the radius $r = 1$. As shown on Fig. 25-5*b*,

For crank 1: $l_1 = r \cos 20° = \ldots\ldots\ldots\ldots\ldots 0.940$
For crank 2: $l_2 = r \cos 260° = -\cos 80° = -0.174$
For crank 3: $l_3 = r \cos 140° = -\cos 40° = \underline{-0.766}$
Sum $= 0.000$

This shows that the first harmonics are balanced.

For the second harmonics the angles α, β, and γ must be doubled and the analysis illustrated by Fig. 25-5*c* gives:

For crank 1: $l_1 = r \cos (20 \times 2) = \cos 40° = \ldots\ldots 0.766$
For crank 2: $l_2 = r \cos (260 \times 2) = -\cos 20° = -0.940$
For crank 3: $l_3 = r \cos (140 \times 2) = \cos 80° = \ldots\ldots \underline{0.174}$
Sum $= 0.000$

The second harmonics are also balanced.

For the fourth harmonics the angles α, β, and γ must be multiplied by 4; Fig. 25-5*d* gives:

For crank 1: $l_1 = r \cos (20 \times 4) = \cos 80° = \ldots\ldots 0.174$
For crank 2: $l_2 = r \cos (260 \times 4) = \cos 40° = \ldots\ldots 0.766$
For crank 3: $l_3 = r \cos (140 \times 4) = -\cos 20° = \underline{-0.940}$
Sum $= 0.000$

The fourth harmonics are also balanced.

For the sixth harmonics the angles must be multiplied by 6, and Fig. 25-5*e* gives:

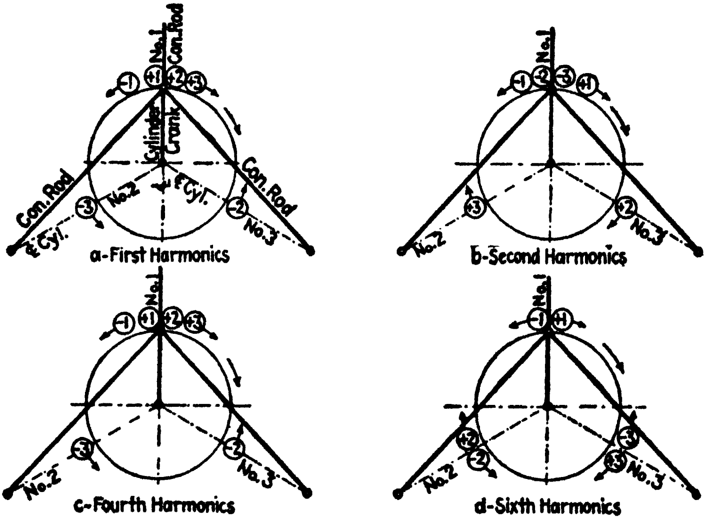
For crank 1: $l_1 = r \cos (20 \times 6) = -\cos 60° = -0.500$
For crank 2: $l_2 = r \cos (260 \times 6) = -\cos 60° = -0.500$
For crank 3: $l_3 = r \cos (140 \times 6) = -\cos 60° = \underline{-0.500}$
Sum $= -1.500$

Thus the first, second, and fourth harmonics of the inertia forces are balanced, but the sixth harmonics are completely unbalanced.

These unbalanced harmonics tend to set up a vibration in the plane of the center lines of the cylinder. The magnitude of these forces, however, is so small that in most cases they may be neglected as far as vibration is concerned.

An analysis for other forces set up by the first harmonics shows, Fig. 25-5*b*, that there exists in the vertical plane an upward resultant from crank 2 and a downward resultant from crank 3. These forces will give a couple in the vertical plane. As the crankshaft turns, the couple decreases, passes through zero, and begins to act in the opposite direction, reaching a maximum at 180° from the initial position. Similar couples will be found in the horizontal plane, showing that a three-crank engine, although its forces of inertia are inherently balanced in the first three harmonics, is not free from vibration because of unbalanced couples.

Six-cylinder Engine.—If a six-cylinder engine is built as a combination of two symmetrically arranged three-cylinder engines, it will have the same inherent balance of the forces of inertia of the first three harmonics, and by virtue of symmetry the couples of each three-cylinder engine will act in opposite directions and will balance each other. The only resulting stresses will be set up in the crankshaft and crankcase and necessitate the use of rigid constructions.



a-First Harmonics

b-Second Harmonics

c-Fourth Harmonics

d-Sixth Harmonics

FIG. 25-6.—Balance analysis of a three-cylinder radial engine.

Radial Engine.—A three-cylinder radial engine has only one crank, and for the first harmonics the three weights rotating with the crank are located all the time at the crankpin, Fig. 25-6*a*. The location of the three imaginary weights rotating counterclockwise can be found by following the same general rule: For the sake of simplifying the analysis the angle may be assumed as zero, $\alpha = 0°$. Then for cylinder 1 at dead center, the weight -1 coincides with the crankpin; for cylinder 2, the crank having moved 120° from its center line, the weight -2 will have moved 120° counterclockwise from the center line of cylinder 2; for cylinder 3 the crank having moved 240° from its center line the weight −3 will be located 240° counterclockwise from the center line of cylinder 3. Thus the weights rotating clockwise can be balanced by a weight equal to one-half of the reciprocating weights of all the cylinders located opposite the crank and at the crank radius. Proceeding in the same manner, the second harmonics of the inertia forces are also found to be half balanced, Fig. 25-6*b*, but these cannot be balanced by a weight fastened to the crankshaft. This is also true for the fourth harmonics, Fig. 25-6*c*. The sixth harmonics are perfectly balanced, Fig. 25-6*d*. As all cylinders of a radial engine are in one plane, no couples are set up.

TABLE 25-1.—INHERENT BALANCE OF DIFFERENT ENGINES

Cylinders		Cranks apart, deg	First harmonics	Second harmonics	Fourth harmonics	Possible to balance
Number	Arrangement					
1	....	....	Unbalanced	Unbalanced	Unbalanced	One-half of first harmonic
2	Parallel	180°	Balanced	Unbalanced	Unbalanced	
2	Opposed	180°	Balanced	Balanced	Balanced	
3	Parallel	120°	Balanced	Balanced	Balanced	
3	Radial	1 crank	One-half balanced	One-half balanced	One-half balanced	Second half of first harmonic
4	Parallel	180°	Balanced	Unbalanced	Unbalanced	
5	Radial	1 crank	One-half balanced	Balanced	One-half balanced	Second half of first harmonic
5	Parallel	72°	Balanced	Balanced	Balanced	
6	Parallel	120°	Balanced	Balanced	Balanced	
6	Parallel	60°	Balanced	Balanced	Balanced	
7	Radial	1 crank	One-half balanced	Balanced	Balanced	Other half of first harmonic
7	Parallel	$51\frac{3}{7}^\circ$	Balanced	Balanced	Balanced	
8	V-type	180°, 4 cranks	Balanced	Unbalanced[a]	Unbalanced[b]	
8	Parallel	90°	Balanced	Balanced	Unbalanced[b]	Up and down unbalance in fourth harmonic
8	V-type	90°	Balanced	Balanced	Unbalanced[b]	
9	Radial	1 crank	One-half balanced	Balanced	Balanced	Other half of first harmonic

[a] $\sqrt{2}$ × unbalanced second harmonics acting horizontally through axis of shaft.

[b] $\sqrt{2}$ × unbalanced fourth harmonics acting horizontally through axis of shaft.

The results of such an analysis of various engines in regard to their inherent balance as to the first, second, and fourth harmonics of the inertia forces and to the possibility of balancing by counterweights part of the unbalanced first harmonics are given in Table 25-1.

25-4. Counterbalancing. *Purposes.*—Counterweights are added to crankshafts for the following purposes:

1. To obtain a static balance of rotating parts.
2. To balance one-half of the first harmonic set up by reciprocating parts.
3. To obtain a dynamic balance.
4. To reduce bearing pressures.

Static Balance.—Weights are added to a crankshaft to make its center of gravity lie on the axis of rotation as in the case of single-cylinder or two-cylinder engines with 360° between the cranks. The weights of the big ends of the connecting rod, which have an essentially rotary motion, must be added to the weights of the crankpin and cheeks to be balanced.

Balancing of First Harmonics.—One-half of an unbalanced first harmonic, set up by the reciprocating parts of a single-throw engine, Table 25-1, may be eliminated by a counterweight attached to the crankshaft opposite the crank and having a momentum *Wr* equal to one-half of the reciprocating weights of each cylinder times the number of cylinders and times the crank radius.

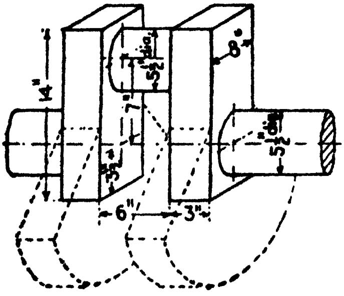


FIG. 25-7.—Balancing crankpin and cheeks.

Example 25-2.—Compute the size of counterweights for a single-cylinder oil engine, 9¼-in. bore, 14-in. stroke, 320 rpm; the crankpin diameter is 5½ in. and its length is 6 in.; the crank cheeks are 8 in. wide and 3 in. thick, Fig. 25-7; the weight of the reciprocating parts is 520 lb; the weight of the large end of the connecting rod is 120 lb.

The weight of the unbalanced ends of the crank cheeks is from Fig. 25-7,

$$3 \times 8 \times 7 \times 0.283 \times 2 = 95.1 \text{ lb}$$

With the dimensions given the center of gravity happens to coincide with the center of gravity of the crankpin. The weight of the crankpin is

$$0.7854 \times (5.5)^2 \times 6 \times 0.283 = 40.3 \text{ lb}$$

Thus the revolving weights to be balanced are equal to

Cheeks	95.1 lb
Crankpin	40.3
Big end of connecting rod	120.0
Total	255.4 lb

In addition one-half of the first harmonics set up by the reciprocating weights can also be balanced, making the total weight

$$255.4 + 520 \times 0.5 = 515.4 \text{ lb}$$

This weight will be divided between two counterweights, 257.7 lb each, if their center of gravity is at the crankpin circle. If the crankcase space limitations force the center of gravity to move closer to the shaft axis, making it 6 in., then the actual weight of the counterweights must be changed in inverse proportion to the distance from the axis of rotation or

$$257.7 \times 7/6 = 300 \text{ lb each}$$

Dynamic Balance.—In this case weights are added to eliminate unbalanced couples or to shift the center of gravity along the axis of rotation. A rotating body is dynamically balanced if its mass is distributed symmetrically about the axis of rotation. A body dynamically balanced is also in static balance, but the converse is not necessarily true. A two-throw crankshaft with 180° between the cranks, Fig. 25-8, if properly machined, is balanced statically but not dynamically. The central double crank A and the center parts of the cranks up to the planes f-g and h-i are in complete balance, but the crankpins B_1, B_2 with the adjacent cheek ends will set up unbalanced centrifugal forces in the planes of rotation. These conditions are shown in Fig. 25-9. The centrifugal force

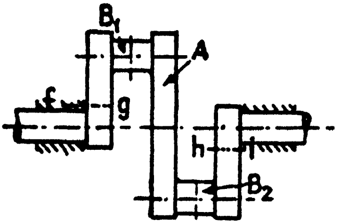


FIG. 25-8.—Two-throw crankshaft.

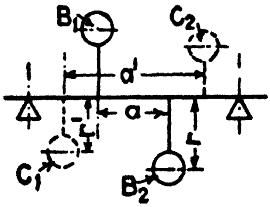


FIG. 25-9.—Dynamic balance.

$$F = mv^2/r = 0.00034Wrn^2 \tag{25-2}$$

where W is the weight of the unbalanced parts, B_1 or B_2, lb,

v is the circumferential velocity of the center of gravity of these parts, fps,

n is the number of revolutions per minute,

r is the radius, ft.

The two centrifugal forces give a couple Fa, where a is the distance between the forces, Fig. 25-9. If an equal but opposite couple lying in the same plane is provided, the system will become dynamically balanced. From expression (25-2) the relation between these two couples is evidently

$$Wra = W'r'a' \tag{25-3}$$

If the counterweights C_1 and C_2 are located as shown in Fig. 25-9, the centrifugal forces will set up certain bending stresses in the crankshaft which can be avoided if the counterweights are placed in the planes of rotation of the unbalanced crankpins.

Example 25-3.—Find the unbalanced couple of a shaft as shown in Figs. 25-8 and 25-9 if the unbalanced weights B_1, B_2 = 185 lb each, a = 16 in., r = 6 in., and engine speed is 600 rpm.

By expression (25-2) the centrifugal force is

$$F = 0.00034 \times 185 \times 6 \times 600^2/12 = 11{,}320 \text{ lb}$$

The unbalanced couple is

$$11{,}320 \times 16/12 = 15{,}093 \text{ lb-ft}$$

Example 25-4.—Determine the weights necessary to obtain a dynamic balance in the foregoing example if the distance a' = 18 in. and r' = 5 in. From expression (25-3),

$$W' = Wra/r'a' = 185 \times 6 \times 16/(18 \times 5) = 197.4 \text{ lb}$$

Reduction of bearing pressures may be desired for engines with crankshafts in dynamic balance, such as vertical engines with four or more cranks. In these engines the centrifugal forces of the big ends of the connecting rods produce a periodical increase of the main bearing pressure. These centrifugal forces can be eliminated by counterbalancing each lower connecting-rod end with a weight located opposite to the crankpin and having an equal momentum Wr.

Example 25-5.—Determine the increase of bearing pressure by the unbalanced weights in Example 25-2.

The centrifugal force due to 515.4 lb is by expression (25-2)

$$F = 0.00034 \times 515.4 \times 7 \times 320^2/12 = 10{,}460 \text{ lb}$$

If the length l of each main bearing is 1.5 times its diameter, one bearing on each side of a crank, the specific pressure will be

$$p = F/(ld \times 2) = 10{,}460/(5.5 \times 1.5 \times 5.5 \times 2) = 115.3 \text{ psi}$$

The allowable bearing pressure being about 500 psi, this means an increase of 23 per cent and shows why counterweights are useful even on multicylinder engines.

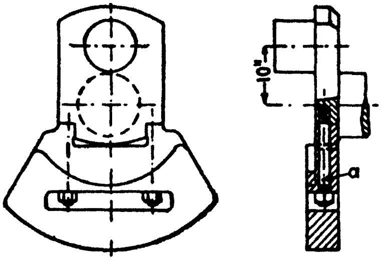


FIG. 25-10.—Gas-engine crank with counterweight.

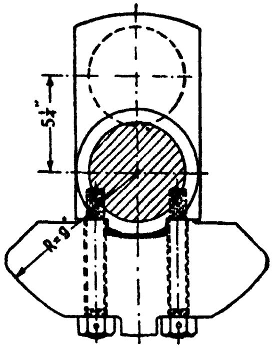


FIG. 25-11.—Counterweight of a compression-ignition oil engine. (*Courtesy of Cooper-Bessemer Corporation.*)

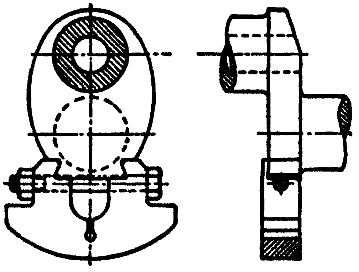


FIG. 25-12.—Counterweight for a gasoline-engine shaft.

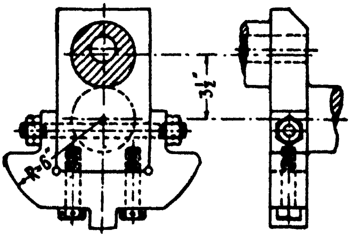


FIG. 25-13.—Counterweight for a high-speed heavy-duty oil engine.

Construction Details.—In small engines and many special constructions the counterweights are made integral with the shaft. In low-speed engines cast-iron counterweights are fastened to the shaft by studs *a*, Fig. 25-10. To prevent any movement of the counterweights on the shaft they are machined to fit over the cheeks but are made slightly smaller. The counterweights are heated before assembling, which gives a light shrink fit.

A similar construction used for a high-speed oil engine is shown on Fig. 25-11. Another method of fastening is shown on Fig. 25-12. A combination of both methods is used in the Winton oil engines, Fig. 25-13.

Figure 25-14 shows the method of fastening a steel counterweight to an extended crank cheek in an airplane engine. The weight can easily be taken off for balance purposes and is held more by friction created by the four cap screws than by the latter in shear.

Figure 25-15 shows a counterweight incorporated in a cast-steel crank-and-pin, shrunk on the end of a forged shaft. Such side crankshafts are used on large horizontal gas engines.

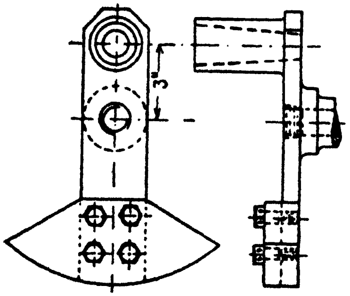


FIG. 25-14.—Crank with counterweight of a radial engine.

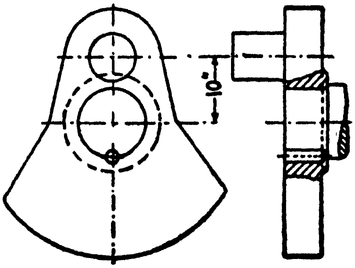


FIG. 25-15.—Crank-counterweight casting.

25-5. Balancing of Harmonics.—*Complete balance of first harmonics* of reciprocating parts can be obtained by using two counterweights, each equal to one-half of the weight of the reciprocating masses, one rotating clockwise and the other counterclockwise. Such an arrangement is used in the single-cylinder 4¼-in. × 5-in. General Motors two-stroke engine running at 1200 rpm. This engine has four balance weights, each equal to one-fourth of the weight of the reciprocating parts, two fastened to the ends of the crankshaft and two to the ends of a special balancer shaft located symmetrically to the other side of the cylinder center line and rotating in the opposite direction to the crankshaft. When the piston is at the top dead center all four weights are in their lowest position.

Balancing of Second Harmonics.—In a four-cylinder engine with cranks at 180° the second unbalanced harmonic, which is set up by reciprocating weights, can be eliminated by setting up equal and opposite forces. This principle is applied in the Lanchester vibration damper, which consists of two disks, each having one side considerably heavier than the other. One of the unbalanced disks is geared to the crankshaft and to the other disk so that they rotate in opposite directions with a speed of twice the crankshaft speed. Their unbalanced weights are such as to counterbalance the second harmonics of the reciprocating parts.

25-6. Synchronous Vibration.—*Natural period* of vibration is the time during which a body or system, set into free vibration or oscillation, will complete one cycle. The reciprocal of the natural period is called *natural frequency f* and is the number of oscillations per second. The natural period of a system depends upon the mass of the system and the force necessary to produce a unit deflection

$$t = 1/f = 2\pi \sqrt{W/12Fg} = 2\pi \sqrt{y/12g} \qquad (25\text{-}4)$$

where t is the natural period, sec,
W is the weight of the oscillating system, lb,
F is the force of deflection, lb/in.,
g is acceleration of gravity, 32.2 fpsps,
y is the deflection by the load W, in.

If there is an unbalanced force, as from the weight of reciprocating parts when an engine is running, and the force application has a time period equal to the natural period of some part of the engine, that part may develop a synchronous vibration.

Critical Speed.—Through resonance, or accumulation of impulses, the amplitude of a synchronous vibration may exceed by many times the deflection which can be produced statically by the unbalanced force, and thus become dangerous. The engine speed at which synchronous vibration of a part occurs is called the *critical speed* in respect to that part. Engine parts that are apt to develop synchronous vibration, "pick up vibration," are push rods, beams, or brackets attached to an engine, and the whole structure in automobiles and airplanes. Valve springs subject to periodical forces also can develop deflections interfering with their proper functioning.

Synchronous vibrations can be set up not only by the first harmonics of unbalanced forces, but also by the higher harmonics, because the frequency of the first harmonics at a certain speed is equal to the frequency of the second harmonics at half the speed, to the frequency of the fourth harmonics at one-fourth the speed, etc. In addition to having a main critical speed, a body may develop synchronous vibrations at one-half, one-fourth, or even one-twelfth of the main critical speed, which in this case also becomes a critical speed. However, critical speeds set up by harmonics of higher orders, having smaller amplitudes, meaning smaller acting forces, gradually become less noticeable. In order to avoid critical speeds during operation of an engine, the natural frequency of vibration of any part should be above that of the lowest engine harmonic at the maximum engine speed.

Engine Supports.—In automotive engines the range of speeds is large, and it is very difficult, usually impossible, to avoid vibration at all speeds, and often there are several critical speeds. In most motor vehicles the transmissions of vibration is stopped by inserting flexible members, such as rubber pads or springs, between the engine and the chassis.

In motor boats of medium size equipped with low-speed engines, the conditions are reversed; usually the mass of the hull is large enough to absorb the vibrations set up by the oil engine, but the engine foundation is not heavy enough to prevent a noticeable vibration of the tall cylinders. This vibration is stopped by connecting the cylinder heads of the first and last cylinder to the hull by sufficiently heavy steel braces.

In airplanes a comparatively small range of engine speeds is used during flight. Therefore it is possible to build the framework so that it will not pick up vibrations from the engine. The natural frequency of vibration of the various sections should be such that resonance cannot occur at normal engine speeds. However, to be on the safe side, the engine mounting is fastened to the aircraft structure by special rubber-cushioned sockets.

Example 25-6.—A motorcycle engine with a maximum speed of 4000 rpm caused excessive vibration of the footboards, fastened to the engine supports, at a speed of 3300 rpm. The engine supports consisted of two steel tubes $1\frac{1}{8}$ in. outside diameter, No. 11 B.W.G., and 19 in. long. The weight of the engine was 30 lb. Find how to eliminate this vibration.

The frequency of vibration can be expressed from formula (25-4) as

$$f = (\sqrt{12g/y})/2\pi \qquad (25\text{-}5)$$

The deflection y of a beam depends upon the method of fastening. For a beam fixed at the ends and loaded at the center

$$y = Wl^3/192\,EI \qquad (25\text{-}6$$

For a beam supported at the ends the numerical coefficient becomes only 48. As the ends of the tubes are fastened to the cycle frame, but the whole construction is not very rigid, it seems justifiable to take a coefficient of 96 and disregard the weight of the tubes. The thickness of the tube wall being 0.120 in., the inside diameter

$$d_0 = 1.125 - 0.120 \times 2 = 0.885$$

and the moment of inertia

$$\begin{aligned} I &= \pi(d^4 - d_0^4)/64 \qquad (25\text{-}7) \\ &= \pi(1.125^4 - 0.885^4)/64 = 0.0486 \text{ in.}^4 \end{aligned}$$

Using for steel $E = 30{,}000{,}000$ and taking into consideration the fact that $W = 30$ lb is supported by two tubes, the deflection becomes

$$y = 30 \times 19^3/(96 \times 30{,}000{,}000 \times 0.0486 \times 2) = 0.00073 \text{ in.}$$

and

$$f = (\sqrt{12 \times 32.2/0.00073})/2\pi = 115.5 \text{ vib/sec}$$

The speed of 3300 rpm is equal to 55 revolutions per second, and the frequency of the second harmonic is 55 × 2 = 110 vibrations per second. Probably the weight of the footboards fastened to the tubes reduced their frequency of vibration from 115.5 to 110 vibrations per second which resulted in a resonance at 3300 rpm.

When increasing the frequency of vibration it is advisable to raise it above the maximum engine speed of 4000 rpm, which is 21 per cent higher than the resonance speed. An increase of 45 per cent will bring the frequency of vibration above the second harmonic at maximum speed. From expressions (25-5) and (25-6), $f_2 = 1.45f$, if $I_2 = (1.45)^2 I$. Keeping the same wall thickness and designating the new outside diameter of the tube by d_2, the latter can be found from expression (25-7) as

$$(1.45)^2 \times 0.0486 = \pi[d_2^4 - (d_2 - 0.240)^4]/64$$

which gives $d_2 = 1.41$ in. or a 1½-in. tube. It may be mentioned that an increase of the thickness of the tube wall, keeping the same outside diameter, would not give the required moment of inertia even with a solid steel rod.

Valve Springs.—If the speed of the camshaft is equal to the natural frequency of the valve spring, or one-half, one-third, one-fourth, etc., of it, there is a tendency for the spring to vibrate with its natural period. Under these conditions the center coils will surge back and forth. This motion changes the spring force and interferes with the proper closing of the valves. Half speed means that the impulse producing the surge is applied every second cycle of spring vibration, one-third speed every third cycle, etc. Evidently the amplitudes produced by these less frequent impulses gradually decrease. Therefore the farther the camshaft speed is from the natural frequency of the spring, the less danger there is of having the surge interference. For this reason it is desirable to use springs with a high natural frequency of vibration. Another method is to use two concentric springs having different natural frequencies, since the tendency of one spring to surge would be damped out by the other spring. To avoid interlocking, one spring should be wound right hand and the other left hand.

The natural frequency of a spring is expressed by the same formula (25-4). However, only one end of the spring has a full movement; the movement gradually decreases toward the other end, which is still. Therefore it can be assumed that only one-half of the spring weight W participates in the movement, which gives from formula (25-4)

$$f = \frac{1}{2\pi}\sqrt{12F \times 32.2 \times 2/W} = 4.42\sqrt{F/W} \qquad (25\text{-}8)$$

where F is the force in pounds necessary to compress the spring 1 in., called the *scale* of the spring.

Example 25-7.—Determine the dangerous engine speeds when using the following valve spring: mean diameter of the spring is 2 in., diameter of wire is ¼ in., eight coils; with the valve closed the spring is compressed 2 in. and has a force of 160 lb; the valve lift is ½ in.

The scale of the spring $F = 160/2 = 80$ lb.

The weight of the spring

$$W = 0.7854 \times (0.25)^2 \times \pi \times 2 \times 8 \times 0.28 = 0.69 \text{ lb}$$

and $f = 4.42\sqrt{80/0.69} = 47.6$ vib/sec $= 2860$ vib/min.

Dangerous engine speeds would be: $2860 \times 2 = 5720$ rpm with a camshaft speed of 2860 rpm; next would be $5720/2 = 2860$ rpm; $5720/3 = 1907$ rpm; $5720/4 = 1430$ rpm. Lower engine speeds will not affect the spring at all.

25-7. Torsional Vibration.—If a couple twists a shaft, it creates a stress which, when the couple ceases to act, starts to bring the shaft back to its original position, but owing to the mass of the shaft the latter will pass it and twist in the opposite direction, creating a new restoring stress and so on until these movements back and forth through the equilibrium position are damped out by internal friction.

Such movements of one section relative to another section are called *torsional vibration*. The forces which produce torsional vibration in internal-combustion engine shafts come from the variable gas pressures on the pistons and from the forces of inertia. A shaft has a natural period of torsional vibration which is a function of its dimensions, characteristics of its material, and masses attached to it. If the acting forces come at regular time intervals and their frequency f_f is a multiple of the natural frequency f_s of the shaft, meaning f_s/f_f is a whole number, a condition of resonance will exist. The smaller the number expressing the ratio f_s/f_f, the more often will the impulse of the disturbing forces occur and the greater will be the resulting amplitude of vibrating. The maximum amplitude of vibration occurring at a critical speed n_c, Fig. 25-16, would become destructive if its increase were not stopped by damping forces.

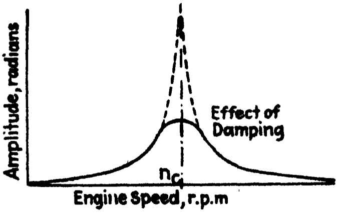


FIG. 25-16.—Effect of damping on vibration.

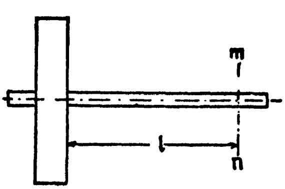


FIG. 25-17.—Node location with one mass.

Node.—In a shaft subjected to torsional vibration there is always one section which does not oscillate and in which the torsional stresses reach their maximum value. This section is called the *nodal section* or simply the *node*, and the amounts of the deflection (angular) are measured in reference to this node. The location of a node depends upon the inertia masses causing torsional vibration. If there is only one rotating mass (flywheel) fastened to one end of the shaft, then the shaft will have only one node, section *mn*, Fig. 25-17, where the farthest of the tangential forces acts, counting from the flywheel. In the case of several inertia masses, as in a multicylinder crankshaft, where the inertia masses of each crank with part of the connecting rod must be taken into consideration, the node will be located somewhere between the first and last of the rotating masses. Its location

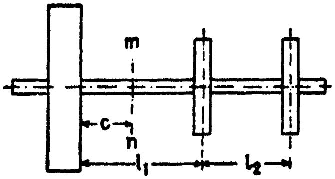


FIG. 25-18.—Node location with several masses.

can be found from the equation of dynamic equilibrium, considering the flywheel effects of all rotating masses as though they were weights and locating the node at their center of gravity. For three rotating masses, a flywheel and two cranks, Fig. 25-18, the equation of equilibrium is

$$I_1c = I_2(l_1 - c) + I_3(l_1 + l_2 - c) \tag{25-9}$$

and solving for c, the distance of the node from one of the ends, gives

$$c = \frac{I_2l_1 + I_3(l_1 + l_2)}{I_1 + I_2 + I_3} \tag{25-10}$$

Both equation (25-9) and Fig. 25-18 mean that a node divides a shaft into two parts, each behaving as a shaft with one oscillating mass, Fig. 25-17.

Frequency.—The natural period of torsional or angular vibration can be determined from a formula similar to formula (25-4),

$$t = 2\pi \sqrt{I/T} = 2\pi \sqrt{Wr^2l/gGI_p} \tag{25-11}$$

where I is the polar moment of inertia of the rotating mass $= Wr^2/g$, in.-lb-sec^2,
T is the shaft constant, or torque, necessary to produce an angle of twist of 1 radian in a shaft 1 in. long, in.-lb,
W is the weight of the rotating mass, lb,
r is the radius of gyration of this mass, in.,
l is the oscillating length of the shaft, in.,
g is acceleration due to gravity, 386 in.psps,
G is the modulus of elasticity in shear (torsion) of the shaft material, psi,
I_p is the polar moment of inertia of the shaft, in.^{4}.

When applying formula (25-11) to a shaft with several rotating masses the length l must be considered as the distance from the node to the application of the inertia mass Wr^2/g, and the frequency per minute may be expressed as

$$f = 60(\sqrt{gGI_p/\Sigma(Wr^2l)})/2\pi = 187.6 \sqrt{GI_p/\Sigma(Wr^2l)} \tag{25-12}$$

Equivalent Shaft.—A shaft of variable cross section may be replaced by an equivalent shaft of a constant cross section. Any portion of a shaft of the length l and diameter d can be replaced, without changing the angle of torsion of the shaft, by a portion of the length l_0 and diameter d_0, where

$$l_0 = l(d_0/d)^4 \tag{25-13}$$

In the case of crankshafts the length of an equivalent shaft depends on conditions of constraint at the bearings. Small bearing clearances give a more rigid shaft and a shorter equivalent shaft. However, the numerical influence of bearing clearances is rather indefinite and not very great, and for many purposes a crankshaft may be considered as a straight shaft with a diameter equal to the diameters of the journals and crankpins. The influence of cheeks also may be neglected, as a first approximation.

If a crankshaft is rigidly connected to other shafts with generator rotors, pump impellers, or ship propellers, it cannot be investigated alone but only as a part of the whole system of interconnected shafts and rotating masses.

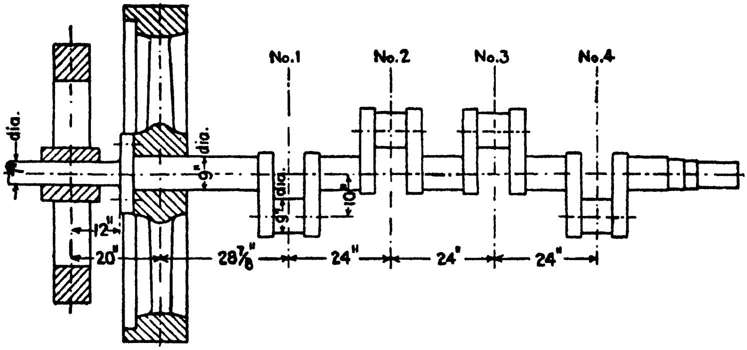


Fig. 25-19.—Four-cylinder oil-engine crankshaft.

Example 25-8.—Analyze the crankshaft of a four-cylinder compression-ignition oil engine, Fig. 25-19, in respect to torsional vibration. Additional engine data are:

Engine speed	240	rpm
Weight of the revolving parts on each crank	650	lb
Weight of the reciprocating parts on each crank	700	lb
Weight of the flywheel	6100	lb
Weight of the generator rotor	2425	lb
Radius of gyration of the cranks	10	in.
Radius of gyration of the flywheel	34	in.
Radius of gyration of the rotor	25.6	in.

The journals and crankpins being of the same diameter, the crankshaft will be considered as a straight shaft 9 in. in diameter. The equivalent length of the 12 in. long 7 in. diameter part of the extension shaft is, according to formula (25-13),

$$l_0 = 12 \times (9/7)^4 = 32.75 \text{ in.}$$

The flywheel effects Wr^2 are

Cranks:	$(650 + 700/2) \times 10^2 =$	100,000 lb-in. sq
Flywheel:	$6100 \times 34^2 =$	7,051,600 lb-in. sq
Rotor:	$2425 \times 25.6^2 =$	1,584,000 lb-in. sq

The equivalent lengths of a uniform 9-in. shaft with flywheel effects, in thousands of pound-inch square, are shown in Fig. 25-20, which also indicates that the node of the torsional vibration probably lies between the two main rotating masses, at c in. from the left end.

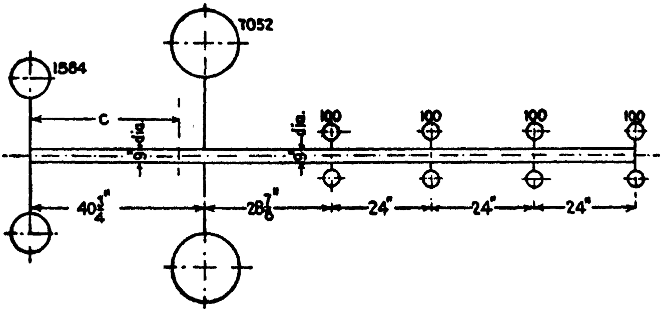


FIG. 25-20.—Diagram of equivalent shaft.

The equilibrium equation can be written similar to equation (25-9)

$$1584c = 7052(40\tfrac{3}{4} - c) + 100(40\tfrac{3}{4} - c + 28\tfrac{7}{8}) + 100(40\tfrac{3}{4} - c + 28\tfrac{7}{8} + 24) + 100(40\tfrac{3}{4} - c + 28\tfrac{7}{8} + 48) + 100(40\tfrac{3}{4} - c + 28\tfrac{7}{8} + 72)$$

which gives $c = 36.5$ in.

The values to be used in formula (25-12) are

$I_p = 0.098d^4 = 0.098 \times 9^4 = 643 \text{ in.}^4$
$Wr^2 = 1{,}584{,}000$ lb-in. sq
$l = c = 36.5$ in.
$G = 11{,}500{,}000$ psi
$f = 187.6\sqrt{11{,}500{,}000 \times 643/1{,}584{,}000 \times 36.5} = 2120$ vib/min

A four-cylinder four-stroke engine has two firing events per revolution; therefore the main critical speed would be $2120/2 = 1060$ rpm. As the engine speed is 240 rpm, only the critical speed of the fourth order, $n = 265$, approaches the normal speed but not very close and, giving comparatively few impulses, should not be dangerous. If the engine is operated on the two-stroke cycle, it would have four firings per revolution and hence the main critical speed would be 530 rpm, the critical speeds of second order 215, and of the third order 177 rpm, which is probably safe although not as good as in a four-stroke engine.

25-8. Elimination of Resonance.—Torsional vibration of a crankshaft is objectionable for two reasons: (1) because it may result in dangerously high stresses in the shaft and (2) because it affects other parts, particularly gears driving the crankshaft, pumps, or blower. Torsional vibration of a crankshaft can be either prevented by a proper design or eliminated by using vibration dampers.

Prevention of Resonance.—To move the critical speeds farther away from the engine speed the natural frequency of torsional vibration of a shaft may be increased by any one or all of the following methods:

1. Increasing the diameter of the shaft.
2. Decreasing the length of the shaft.
3. Increasing the rigidity of the shaft by adding bearings.
4. Decreasing the weight of the flywheel.

The first method can be combined with that of making the shaft hollow. The polar moment of inertia of a hollow shaft with an outside diameter d_1 and inside diameter d_2 being

$$I_p = \pi(d_1^4 - d_2^4)/32 \qquad (25\text{-}14)$$

it is evident that the taking out of even a considerable part at the center has little effect on the rigidity of the shaft. For example, a hollow shaft with $d_2 = 0.3d_1$ has a moment of inertia only 0.81 per cent smaller but a weight 9 per cent smaller than a solid shaft.

The second method can be applied to a crankshaft connected to a long extension shaft, such as a propeller shaft in a boat or in an aircraft. By inserting a flexible coupling between the crankshaft and driven shaft the length of each one is decreased and the rigidity increased.

The third method can be applied to automobile engines in which the cranks do not have bearings on each side: four-cylinder crankshafts with two or three bearings; six-cylinder crankshafts with three or four bearings; etc. Adding bearings to such a crankshaft increases its rigidity and raises the natural frequency of vibration.

The fourth method is evident from formula (25-12).

Vibration dampers are used to reduce the amplitude of torsional vibration without changing the natural frequency of the shaft. Internal friction in the shaft material and friction of the oil film in the bearings acts as a damper, but in some cases this may not be sufficient. By causing vibrations to do work they can be damped effectively. A vibration damper used in automobile engines is shown in Fig. 25-21: A two-piece flywheel a floats pressed by springs c to the disk b which is lined with friction disks f and fastened to the engine shaft. When the shaft and with it the disk b begin to oscillate, the friction between b and a damps down the vibration and prevents it from becoming objectionable.

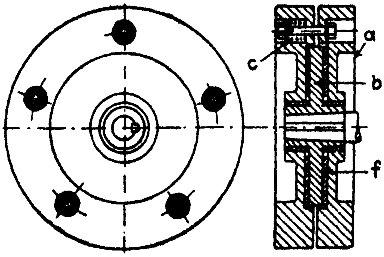


FIG. 25-21.—Vibration damper.

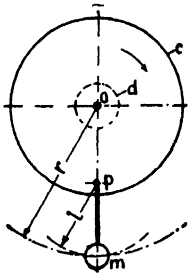


FIG. 25-22.—Action of a pendulum torsional-vibration damper.

Pendulum Damper.—This damper consists of a mass m, Fig. 25-22, suspended from point p of a disk c, which is fastened to and rotates with the shaft d. If the rotation is sufficiently fast and uniform, m rotates, owing to centrifugal force, about o on a radius r. If the shaft d is subjected to torsional vibration, the angular velocity of the disk c will fluctuate and m will begin to swing about p as a simple pendulum with a length l. The frequency of such a simple pendulum is, in vib/sec,

$$f = \sqrt{g/l}/2\pi \qquad (25\text{-}15)$$

where g is acceleration due to gravity,
l is the pendulum length.

If the whole system rotates with a speed as used in engines and the radius r is not too small, the influence of the force of gravity upon the pendulum can be disregarded. In equation

(25-15), instead of g, the radial acceleration due to rotation must be used

$$a = v^2/r = 4\pi^2 n^2 r \tag{25-16}$$

This gives for the frequency

$$f = n\sqrt{r/l} \tag{25-17}$$

which shows that the natural frequency of swinging of a rotating pendulum is directly proportional to the engine speed and is determined by the dimensions r and l. If the frequency of the pendulum is made equal, or very near to, or a multiple of the frequency of a certain harmonic of torsional vibration of the shaft, which appears at a certain critical speed, keeping the pendulum swinging will absorb some of the energy of the system and thus damp down that particular harmonic.

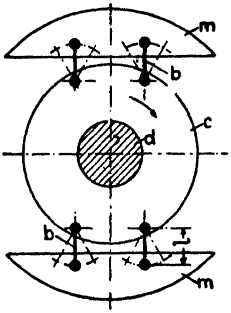


FIG. 25-23.—Schematical arrangement of a torsional-vibration pendulum damper.

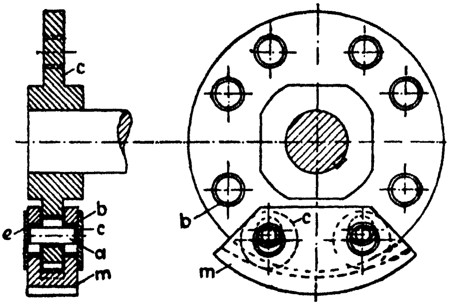


FIG. 25-24.—Torsional-vibration pendulum damper.

The mass of the pendulum does not influence the frequency of the pendulum but influences only the amount of energy absorbed. The heavier the pendulum, the greater is the energy absorbed and the more effective is the damper.

A damper based on the pendulum principle is shown in Fig. 25-23.[1] Two masses m,m are suspended by links b, b to a disk c; the disk is fastened to the free end of the crankshaft, the one which has the maximum amplitude of torsional vibration. When the shaft begins to vibrate torsionally, the masses m,m begin to swing and absorb part of the torsional energy. By changing the length l of the links the frequency f, equation (25-17), can be adjusted to make it equal to the frequency of the dangerous harmonic. The masses are used in pairs in order not to disturb the shaft balance.

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[1] TAYLOR, E. S., Crankshaft Torsional Vibration in Radial Aircraft Engines, *SAE Journal*, vol. 39, p. 81, 1936.

In practice the weights m are suspended not by links but by large-diameter pins a,a of hardened steel inserted in oversize holes with hardened bushings b,b in the disk c and the weights m, Fig. 25-24. The pins are held in place by retainer plates e,e. Figure 25-24 shows a damper with four weights, of which three are taken off. Using pins of a slightly different diameter is equivalent to changing the length of the links. Thus the damper can be "tuned" to a desired frequency. If the shaft has two critical speeds, one pair of weights is tuned to one critical speed, and the second pair, at right angles to the first pair, is tuned to the second critical speed. The damping effect can be intensified by increasing the weight of the swinging masses and the distance of their center of gravity from the axis of rotation. These dampers are used rather extensively in high-speed multicylinder engines.

Counterweights attached to the cheeks of a crankshaft can be made to act also as dampers by suspending them on pins, as in Fig. 25-24, instead of fastening them rigidly.

Detuner Flywheel.—A special type of damper is used in the Doxford opposed-piston marine oil engines.[1] The crankshaft has two medium-weight flywheels, one on each end; the flywheel on the free end is connected to the shaft by means of a flexible spring coupling, with a spring rigidity increasing with the angle of deflection. This gives a variable natural frequency of vibration to the whole shaft system and prevents exact synchronism to be established at any speed; hence the name detuner.

25-9. Foundations. *Purposes.*—Generally speaking the foundation of a reciprocating engine must perform three functions: support the weight of the engine, maintain the engine in proper alignment with the driven mechanism, and absorb vibrations produced by unbalanced forces due to reciprocating and revolving masses. The third function is the most important, but if an engine is properly balanced, as are automobile and airplane engines, it does not need a foundation, *i.e.*, not a heavy one of concrete.

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[1] *Engineering* (London), vol. 155, No. 4025, p. 182, Mar. 5, 1943.

Methods.—The first object is achieved by giving the foundation a sufficient supporting area. The safe loads vary from about 1000 psf for alluvial soil or wet clay, up to 4000 psf for gravel, coarse sand, or dry clay. In these computations, 135 lb/cu ft can be used for the weight of concrete.

The second object is attained by making the foundation strong and rigid, reinforcing it with steel bars in larger sizes, especially if the soil is not firm. In the case of a belt drive, there must be a sufficient side area and weight to resist the outside forces. A practical rule is to make the foundation depth equal to from 3.2 to 4.2 times the engine stroke.

The third object is achieved by giving the foundation a sufficient mass or weight. Theoretically the minimum weight required to absorb vibrations could be expressed as a function of the weight of reciprocating masses and engine speed. Practically this is not done, and a simple empirical formula is used, namely,

$$W_f = CW_e\sqrt{n} \tag{25-18}$$

where W_f is the weight of the foundation lb,
W_e is the weight of the engine, lb,
C is an empirical coefficient,
n is the engine speed, rpm.
The coefficient C is given in Table 25-2.

TABLE 25-2.—VALUES OF COEFFICIENT C IN FORMULA (25-18)

Type of engine	Cylinder arrangement	Number of cylinders	Coefficient C
Single acting............	Vertical	1	0.15
Single acting............	Vertical	2	0.14
Single acting............	Vertical	3	0.12
Single acting............	Vertical	4-6-8	0.11
Single acting............	Horizontal	1	0.25
Single acting............	Horizontal, duplex	2	0.24
Single acting............	Horizontal, twin duplex	4	0.23
Double acting...........	Horizontal	1-2	0.32
Double acting...........	Horizontal, twin tandem	4	0.20

Vibrations.—An average person can detect vibrations of 0.004 in., and more sensitive persons may notice vibrations as small as

0.002 in. The problem of minimizing vibration pertains not so much to the engine and its foundation as to the engine building and sometimes even to other buildings in the neighborhood. Many parts of a building have low natural frequency of vibration and can develop resonance. There exist several measures of precaution against transmission of vibrations: first, to make sure that the engine foundation does not come in direct contact with any part of the building foundation; second, to insulate the engine foundation from the subsoil by putting under it a layer of cork. Specially prepared slabs made of compressed granulated cork can bear a load up to 3500 psf. A third measure is to insulate the engine bedplate from the foundation by putting it on special coil-spring supports, as shown in Fig. 25-25. The supports are placed under the foundation-bolt holes of the bedplate and screws *a* inserted through them; the bedplate is secured to plates *c* by lock nuts *b*, and plates *f* are fastened to the foundation; the weight of the engine is supported by springs *e*.

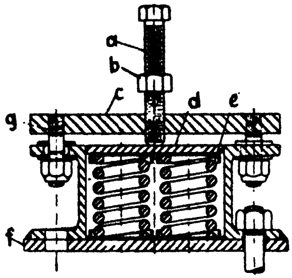

FIG. 25-25.—Vibration isolator.

However, a theoretical analysis[1] shows that a flexible support can be applied only if the number of impulses from an engine is more than 1.5 (better, 4 to 5 times) greater than the natural frequency of the system above the flexible support.

Foundation Bolts.—The size and number of foundation bolts cannot be determined theoretically because an engine bedplate is grouted in on the foundation block, and if properly grouted an engine could run without any bolts. A good practical rule is to make the bolt diameter equal to one-twelfth of the cylinder bore for low-speed engines increasing it up to one-eighth of the bore for high-speed engines. The number of bolts is determined by the bedplate construction. Common practice is to bend the lower end of the bolt to form a hook or ring and to embed the bolts when pouring the concrete. To prevent pulling out of the bolts through tightening of the nuts, the embedded length should be equal at least to 30 bolt diameters.

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[1] SODERBERG, The Vibration Problems in Engineering, *Elec. Jour.*, 1924, pp. 39 and 160.

25-10. Problems.—**1.** Analyze a four-cylinder vertical engine with cranks at 180° in regard to first, second, and fourth harmonics of the inertia forces.

2. Analyze a five-cylinder vertical engine with cranks evenly spaced in regard to the first, second, third, and fourth harmonics, (*a*) graphically and (*b*) analytically.

3. Analyze by computations a seven-cylinder vertical engine with cranks evenly spaced in regard to the first and fourth harmonics of the inertia forces.

4. Analyze a five-cylinder radial engine in regard to first, second, and fourth harmonics.

5. Analyze a six-cylinder V-type engine with a V of 60° and the three cranks at 120° in respect to the first, second, and fourth harmonics of inertia forces.

6. Check analytically an eight-cylinder V-type engine with a V of 60° and four cranks at 180° in respect to the first and third harmonics of inertia forces.

7. Determine the size of the counterweights for a two-cylinder horizontal gas engine with cranks at 360°; the engine has a $12\frac{1}{4}$-in. bore, a 22-in. stroke, and 235 rpm rated speed; the weight of the trunk piston and small end of the connecting rod is 830 lb; large end of connecting rod weighs 215 lb; shaft and crankpin diameter is 6 in.; length of crankpin 8 in.; cheeks are 4 in. thick by 10 in. wide; each of the three main bearings is $8\frac{1}{2}$ in. long.

8. Determine the counterweights necessary to balance dynamically the engine in problem 7 if the cranks are made 180° apart.

9. Determine the size of counterweights which would be required to balance dynamically the engine of problem 7 if the cylinders are opposed, the cranks are 180° apart, and the distance between the cylinder center lines is $24\frac{1}{2}$ in.

10. Determine the action of the unbalanced parts upon the bearing pressures in the engines in problems 7, 8, and 9 before counterweights are installed. Connecting-rod length is $52\frac{1}{2}$ in.

11. Determine the size of the counterweights for a four-stroke three cylinder $4\frac{1}{8}$-in. × $4\frac{3}{4}$-in. radial engine: the rated engine speed is 1800 rpm; the weight of the reciprocating parts is 4.12 lb per cylinder; the cluster with the large ends of the connecting rods weighs 3.21 lb; the crankpin has a diameter of $2\frac{1}{2}$ in. and is $2\frac{7}{8}$ in. long; the cheeks are $\frac{3}{4}$ in. thick and 3 in. wide.

12. Determine the natural frequency of vibration of a standard angle iron with equal legs, 2 in.; thickness $\frac{3}{16}$ in., 4 ft between supports, ends hinged, and loaded only by its own weight.

13. Determine the natural frequency of the angle in problem 12 if the ends were fixed.

14. (*a*) Determine the natural frequency of a helical valve spring: the pitch diameter is 3 in.; diameter of the wire is $\frac{3}{8}$ in.; there are nine free coils and one inactive coil on each end; the initial compression with the valve closed is $1\frac{1}{2}$ in.; the valve lift is $\frac{7}{8}$ in.; for each 100-lb pressure the spring compresses $\frac{15}{16}$ in.; (*b*) determine what engine speed between 200 and 260 rpm may cause the spring to surge; resonance of what order may be feared if the engine is of the four-stroke type.

15. (*a*) Determine the natural frequency of a helical valve spring: the outside diameter of the spring is 1⅝ in.; the diameter of the wire is $\frac{3}{16}$ in.; there are 4¼ active coils and one inactive coil at each end; the free length is $2\frac{7}{16}$ in.; the initial compression is ⅜ in.; the valve lift is ⅝ in.; the spring scale is 110 lb/in.; (*b*) determine resonance of what order may be dangerous if the spring is used on a four-stroke engine operating at 2050 rpm ±5 per cent.

16. A fuel pump of a four-stroke oil engine is returned by a spring: The mean diameter of the spring is 1¼ in., steel wire is No. 10 W. & M. gage; there are twelve free coils and one inactive coil on each end. Find the initial compression of the spring necessary to return the plunger. The engine operates at 1800 rpm; the weight of the plunger is 15 oz, its stroke is ½ in., the time allowed for returning the plunger is a whole revolution. Also determine the natural frequency of vibration of the spring, dangerous engine speeds, and of what order the critical speeds are.

17. Find the equivalent length of a shaft system consisting of an oil-engine crankshaft 7¾ in. diameter, 12 ft 6 in. long; an intermediate shaft 5½ in. diameter, 28 ft long; and a propeller shaft 6 in. diameter, 18 ft long.

18. Find the natural frequency of torsional vibration of the combined shaft of problem 17 if there is, between the crankshaft and intermediate shaft, a flywheel of 2630 lb with a radius of gyration 2.08 ft. The propeller weighs 650 lb and has a radius of gyration of 18 in.

19. Determine the frequency of torsional vibration of a 3-in. steel shaft, 6 ft long, and find the outside and inside diameters and the frequency of vibration of a hollow shaft which can transmit the same torque. The inside diameter of the hollow shaft should be equal to one-half of the outside diameter.

20. Determine the lengths of links of a pendulum damper for the crankshaft of Example 25-8 for two critical speeds: (*a*) of 212 and (*b*) of 265 rpm. The available space does not permit a radius *r* greater than 10 in.

21. A vertical 12-in. × 16-in. 200-hp four-cylinder oil engine operates at 280 rpm and weighs 30,000 lb. Determine the minimum weight of the foundation, its volume, depth, and horizontal area if the safe soil pressure is 2000 psf.

22. Determine the size of the foundation for the engine in problem 7; also size and length of foundation bolts.

23. Compute the horizontal rocking motion of the engine foundation in problem 7 if the counterweights were not installed.

24. Compute the theoretical rocking motions of the engine foundation in problems 8 and 9 if the engine were operated without counterweights and with the space between the concrete and the sides not filled with dirt.

25. A horizontal 13-in. × 18-in., 130-hp duplex gas engine operates at 275 rpm. Determine the minimum foundation weight, volume, depth, and horizontal area. The engine weighs 13 tons, and the safe soil pressure may be assumed as 1500 psf.

26. Determine the number, size, and length of foundation bolts for the engine of problem 24.

CHAPTER 26

LUBRICATION

26-1. General. *Purposes of Lubrication.*—Lubrication as the admittance of oil between two surfaces that are in contact and in a relative motion one to another may accomplish one or several of the following purposes:

1. Reduce friction and wear.
2. Seal a space adjoining the surfaces—such as a piston moving in a cylinder.
3. Cool the surfaces by carrying away heat generated by friction.
4. Clean the surfaces by washing away carbon and metal particles caused by wear.

Of all these objects the most important one is the first one. Lubrication decreases friction by interposing a film of oil between the sliding metal surfaces. If the film is heavy enough to separate the two surfaces so that they will not touch one another, then wear will be eliminated; otherwise it will be only reduced. The thickness of the oil film depends upon the viscosity of the oil and the specific bearing pressure. Viscosity of lubricating oil in turn depends upon the source from which the oil is obtained and the oil temperature. At present, except in special cases, only mineral oils are used for lubricating internal-combustion engines.

The working conditions, such as temperature, bearing pressure, and rubbing speed of different engine parts, vary considerably; and, theoretically, these different parts should be lubricated with different oils. However, the use of several or even of two different kinds of oil in one engine would complicate its design and servicing. Therefore the oil refineries have developed different lubricating oils from which an oil can be selected which is satisfactory for all parts of a certain engine.

Systems of Lubrication.—The methods of lubrication can be divided into three groups—gravity, mechanical without pressure, and pressure systems.

Gravity systems comprise:

1. Drop oilers, individual, as sight-feed oil cups, and common supply tanks with feed tubes to different points.
2. Wick oilers.
3. Hand oiling from cans.

Mechanical systems comprise:

1. Oil-ring lubrication.
2. Splash lubrication.
3. Oil bath as used for gears and chains.
4. Combination of splash and gravity flow.

The main disadvantage of the gravity feed and of some of the mechanical systems is that the quantity of oil flowing to the surfaces that must be lubricated is comparatively small.

Pressure feed systems comprise:

1. Force-feed lubricators with various types of pumps.
2. Banjo lubricators using centrifugal force.

Oil Circulation.—With the exception of a small part, which is burned in the cylinder or leaks out from bearings to the outside, the oil from all lubricated surfaces flows to a low point, acting as an oil sump, from which it is picked up by a so-called *scavenge pump* and returned to the lubricator.

To maintain the lubricating quality of the oil it should be passed on its circuit through a filter which removes the foreign matter, such as particles of metal, carbon, and dust, which gradually contaminate the oil circulating in an engine. After the filter, in order to increase the thickness of the oil film through increased viscosity, the oil on its circuit should be passed through an oil cooler. In stationary and marine engines the oil cooler consists of a heat exchanger with water as the cooling medium. In aircraft engines the oil cooler is made of a coil with air-cooled fins.

Oil Reconditioning.—Oil circulated in an internal-combustion engine does not wear out or break down but becomes unfit for further use through contamination by solid particles and water, through dilution by liquid fuels and oxidation. The water may come from condensed combustion products or from leaks in the cooling system.

One of the simplest and best methods of reconditioning engine lubricating oil is by means of centrifuges. In larger engines the oil is circulated through the centrifuge all the time; in smaller engines batch purification may be used. Proper purification of lubricating oil not only reduces the oil consumption very materially but also increases the life of the engine through decreased wear.

Oil Dilution.—In gasoline engines dilution of the lubricating oil with fuel occurs mainly during starting but also during operation if the fuel is not completely vaporized before it enters the cylinder. After dilution reaches a certain percentage, it does not go further because of evaporation from the hot crankcase oil.

In oil engines dilution with fuel oil occurs while the engine is being turned over before it begins to fire. Dilution may also come during operation through leaks from the fuel system. In properly designed engines dilution is practically absent. If it is present, the sump oil should be watched and when dilution reaches about 5 per cent, the oil should be changed.

Oil Consumption.—The amount of oil actually used varies between 0.00025 and 0.0005 gal/hr per rated brake horsepower in four-stroke oil and gas engines with a properly designed and operated lubrication system. In two-stroke engines the consumption is about twice as great. In air-cooled airplane engines the amount of oil used is much greater, from 0.00085 to 0.002 gal/hr-hp. In good automobile engines the oil consumption is about the same as in stationary engines, but when an automobile engine begins to pump oil, the consumption may become even greater than in airplane engines.

In practice it is customary to express the lubricating-oil consumption by the number of rated horsepower-hours per gallon of oil. The above given figures correspond to about 2000 to 4000 hp-hr/gal, for four-stroke engines and 1000 to 2000 hp-hr/gal, for two-stroke engines. For air-cooled aircraft engines the figures are 500 to 1200 hp-hr/gal.

Importance of Lubrication.—The importance of proper lubrication cannot be overemphasized. Regardless of how well an oil engine is designed from the standpoint of thermal efficiency and strength, and how well it is built from the standpoint of materials and workmanship, if the lubrication is not taken care of properly, the engine will either not run at all or show heavy wear and have a short life.

Friction Surfaces.—The principal friction surfaces requiring lubrication in an internal-combustion engine are:

1. Pistons and cylinders.
2. Crankshaft and main bearings.
3. Crankpins and their bearings.
4. Crossheads and guides.
5. Wristpins and their bearings.
6. Valve gear.

26-2. Pistons and Cylinders. *Splash Lubrication.*—Pistons of high-speed engines, including automobile and aircraft engines, usually do not need special lubricating provisions. The oil which flows from the main and crank bearings and is splashed by the cranks and connecting rods, and the oil mist which is formed in the enclosed crankcase by the fast-moving parts, furnish sufficient lubrication for the pistons. These conditions exist not only when the oil in the crankcase is kept at a certain level to allow its pick-up by the connecting rods but even with a so-called *dry crankcase.*

Positive-feed Oilers.—Pistons of low- and medium-speed engines are lubricated by positive-feed mechanical oilers. Figure 26-1 gives a schematic section of such an oiler: The oil from a supply reservoir *a* drips into the chamber *c*, the number of drops per minute being regulated by a screw *b*; the plunger *d* on its downward stroke, having covered the connection to *c*, pushes the oil into the copper tube *e* which is discharging the oil to the cylinder surface; the check valve *f* prevents the oil from being sucked back by the upward stroke of *d* and also separates the space *c* from the engine cylinder.

In cylinders cast in one piece with the water jacket, special bosses *b* are cast, Fig. 26-2, to take the oil nipple *e*. In cylinders with liners the oil tube is fastened to a nipple *g*, Fig. 26-3, with a conical pipe thread screwed into the liner and a lock nut *h* and rubber ring *k* to seal the water from the outside. To avoid rusting, the nipple *g* is made of brass or monel metal.

Location.—In single-acting engines the hole discharging oil should come between the first and second piston rings when the piston is at the lower or outer dead center, so that the oil hole will never be exposed. In double-acting engines the oil holes are located halfway between the cylinder ends and become exposed at certain piston positions. To prevent the gases from getting into the oil tube a check valve is arranged either in the oil delivery nipple itself or close to it.

To avoid the possibility of discharging the oil during the expansion stroke, when the cylinder is filled with very hot gases, in some engines, in which the oil opening is exposed, the oil delivery is timed and occurs during the very beginning of the compression stroke, when the oil hole is covered by the piston.

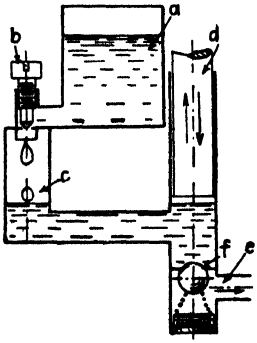

Fig. 26-1.—A mechanical oiler.

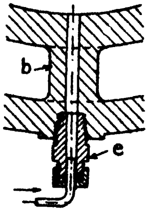

Fig. 26-2.

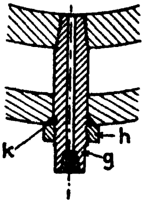

Fig. 26-3.

Cylinder lubrication.

In horizontal cylinders the oil usually is delivered at three points of the cylinder circumference—at the top and 90° from it to each side. In small vertical engines the oil is delivered at two points of the circumference, 180° apart, in a plane normal to the crankshaft center line. In larger vertical engines the oil is delivered in four to eight equally spaced points, all connected to one circuit, Fig. 26-4, to ensure uniform oil pressure and distribution.

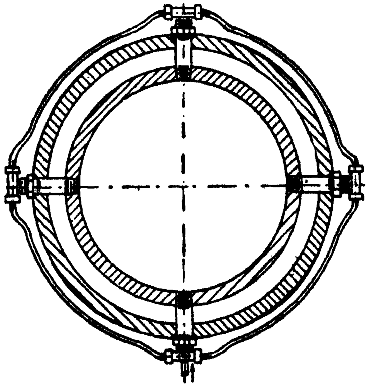

Fig. 26-4.—Cylinder lubrication of larger vertical engines.

Amount of Oil.—The best results are obtained with a minimum amount of oil, just enough to maintain a film of oil on the cylinder surface. Sticking piston rings, much carbon in the circulating oil, and even excessive wear of the liner and piston rings can be traced to excess lubrication of the cylinders. Consumption of cylinder oil varies between 0.0001 and 0.0005 gal/hp-hr, depending upon the engine type and the attention given it by the operators.

Even cylinders of medium-speed engines receive a considerable amount of oil from the crankcase splash and mist, so that it may become necessary to devise scraper rings or vent holes to remove the superfluous oil.

26-3. Crankshafts and Wristpins. *Main Bearings.*—In small stationary engines the main bearings are made with ring oilers. In larger engines with open or at least not very tight crankcases, as in most horizontal engines, the main bearings are lubricated by positive-feed lubricators such as are used for the cylinder lubrication. Engines with enclosed crankcases, as most vertical engines, are usually built with pressure lubrication. The oil is circulated in fairly large quantities, from 0.01 to 0.05 gal/min per rated horsepower. The oil is delivered by means of a gear pump under a pressure of 20 to 60 psi into a manifold, which distributes it to the bearings. Oil flowing out of the bearings goes to an oil sump from which it is picked up by a second, slightly larger scavenge pump, pumped through a filter, then an oil cooler, and to a supply tank from which it flows back to the pressure pump. The oil circulating in the above given quantities and in amounts two to three times as much in some two-stroke engines acts not only as a lubricant but also as a cooling medium for the bearings and, in some designs, for the pistons.

Gear Pump.—The size of a gear pump is determined by the quantity of oil that is displaced by its teeth. The quantity Q in gallons per minute can be computed from

$$Q = 2htbin\eta/231 \tag{26-1}$$

where h is the height of the teeth, in.,
t is their thickness, in.,
b is their width, in.,
i is the number of teeth, the same in both gears,
n is the speed, rpm,
η is the volumetric efficiency, from 0.70 to about 0.85.

Substituting the usual proportions

$$h = 0.687p$$
$$t = 0.5p$$

where p is the circular pitch, gives

$$Q = p^2 b i n \eta / 337 \tag{26-2}$$

The width is made $b = 2p$ to $4p$, and the number of teeth varies from $i = 9$ or 10 in a small engine up to 20 to 24 in a large one.

Pressure Regulation.—Figure 26-5 shows a pressure-regulating valve b, whose spring can be screwed down to maintain the desired pressure. When the bearings are in good shape, the pressure pump delivers more oil than can flow through the small clearances and the oil pressure rises and opens the spring-loaded valve, discharging part of the oil from the pressure line a to the suction line c. As the bearings wear down, their clearances grow, more oil begins to ooze out through the bearings, and less goes to the suction line c. When the bearing clearances increase so much that no oil flows to c, the pressure gage will begin to show a decrease of pressure, and it is time to overhaul the bearings. A spring-loaded valve acts also as a safety valve protecting the whole system from an excessive pressure rise in case one or several discharge points become plugged up.

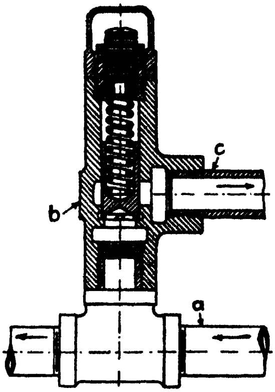

FIG. 26-5.—Pressure-regulating valve.

Oil Piping.—The oil pipes should be installed of such size that the average oil velocities in the suction line do not exceed 10 to 20 fpm and, in the discharge lines, 30 to 40 fpm and about twice that in aircraft engines.

Oil Seals.—To prevent oil leakage along the shaft, collars or oil-throw rings a are put on the shaft, Fig. 26-6, utilizing the increase of centrifual force with the increase of diameter; a hole b drains the oil back into the crankcase. A better seal is obtained by using metal packing rings, Fig. 26-7: b is a split cast-iron collar clamped tightly to the shaft a; d, d are two pairs of bronze or cast-iron snap rings, similar to piston rings, fitted on the sides; c is a cast-iron split housing bolted to the crankcase.

Crankpin Bearings.—In a small engine with enclosed crankcase the crankpins are sometimes lubricated by splash oil. This is safe only if the bearing pressure is low. A better scheme is to have on the bottom of the connecting rod some scoop device that dips into the oil when coming down, Fig. 26-8.

Small horizontal engines and some two-stroke vertical engines use the centrifugal banjo oiler, Fig. 26-9. The oil hole leading to the surface of the crankpin is often drilled at an angle of about 30° in advance of the dead center, so that the upper shell receives oil before the ignition stroke and at a point of relatively low pressure.

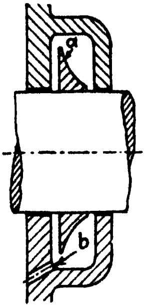

FIG. 26-6.—Spill ring.

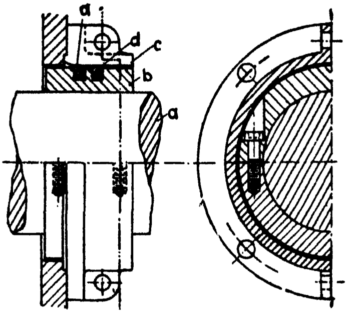

FIG. 26-7.—Crankshaft oil seal.

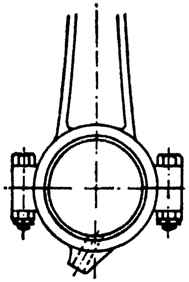

FIG. 26-8.—Connecting-rod end for splash lubrication.

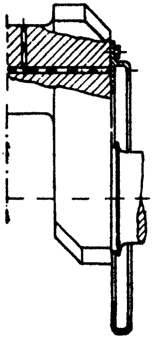

FIG. 26-9.—Banjo oiler.

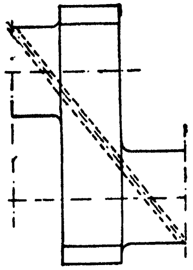

FIG. 26-10.—Drilling of crankshaft for crankpin lubrication.

With pressure-feed lubrication the shaft is drilled either to conduct oil from the adjoining journal to the crankpin, Fig. 26-10, or the whole shaft is drilled and the crankpins receive oil from the inside of the shaft, which is fed by all main bearings, Fig. 28-11.

Crossheads.—Crosshead guides require only a small amount of oil as the specific pressures are usually very low. A line from the force-feed lubricator can take care of the need even in a large engine.

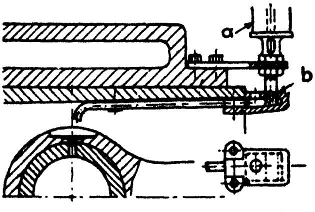

FIG. 26-11.—Wristpin lubrication by a sight-feed oiler.

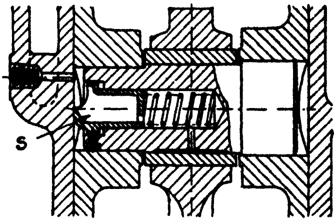

FIG. 26-12.—Wristpin lubrication by a scraper device.

Wristpin Bearings.—In small horizontal engines the wristpin is lubricated by a sight-feed oiler *a*, the oil drops being picked up by a wiper *b*, Fig. 26-11, when the piston comes out from the cylinder.

In small vertical engines the wristpin may be lubricated by oil scraped from the cylinder walls by a special scraper scoop *s*, Fig. 26-12.

In vertical engines, with pressure lubrication delivered by a pump or with banjo-lubricated crankpins, the wristpin is lubricated by excess oil from the crankpin bearing. The oil is delivered either by a hole drilled through the connecting rod, Fig. 26-13 or Fig. 27-25, or, if the connecting rod has an I-section that is too thin to be rifle drilled, by a copper tube fastened with clamps to the inner web of the rod.

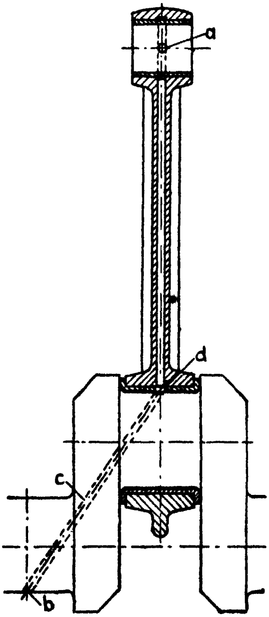

FIG. 26-13.—Pressure-lubrication of crankpin and wristpin.

In computing the pressure of the oil reaching the wristpin, one must notice that a point *a*, Fig. 26-13, just before top center, the oil pressure p_a is equal to the pressure p_b of the pressure system plus the pressure due to the centrifugal force from *c* to *d*, part *b*-*c* being balanced, plus the force of inertia of the column of oil *a-d*, and in a vertical engine, minus the weight of this column.

26-4. Lubrication for Rotary Motion. *Perfect Lubrication.*—According to the hydrodynamic theory of lubrication, if the surface velocity of the journal is high enough and if there is a sufficient oil supply, enough oil pressure may be developed to raise the shaft off the bearing and make it float upon the lubricant. In this case the position of the shaft in the bearing is as shown in Fig. 26-14: The center of the shaft is moved to the left, the point of nearest approach is b', with a minimum oil-film thickness h_m in.; the point of maximum pressure is at c, between the point b and the bottom of the bearing. The rotating journal surface acts as a pump; the oil pressure increases with an increase of the speed.

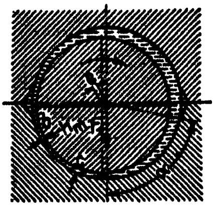

FIG. 26-14.—Oil film between a rotating shaft and its bearing.

Such lubrication, when there is no metal-to-metal contact, is termed *thick-film* or *perfect lubrication.* Because of the clearance which must exist in order to obtain the eccentric position of the shaft necessary for the pumping action, part of the oil is discharged at the ends of the bearing with a corresponding drop of pressure. Figure 26-15 shows the pressure distribution in the oil film in the plane of the maximum pressure along the bearing length for the different loads, 100, 200, 300, and 400 psi, referred to the projected journal area. The oil-film pressure drop toward

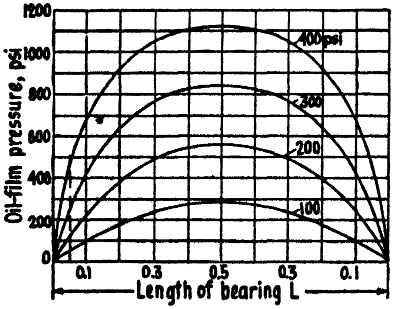

FIG. 26-15.—Longitudinal distribution of oil pressure.

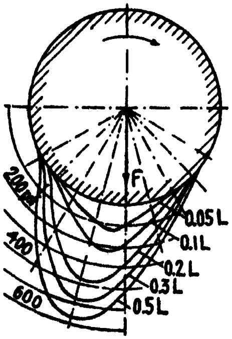

FIG. 26-16.—Oil-pressure distribution around the circumference of a bearing.

the end of the bearing is caused by the oil leaking out at the ends. Typical pressure-distribution curves around the circumference, at different planes of rotation at the same load, 200 psi, are shown in the form of a polar diagram, Fig. 26-16. The curve marked $0.05L$ gives the pressure around the bearing at a distance $0.05L$ from the end of the bearing, L being the length of the bearing; the curve $0.5L$ gives the pressure at the center of the bearing.

Viscosity.—For perfect lubrication, the most important property of the lubricating oil is its viscosity. The unit of absolute viscosity, as used in technical computations, is the *centipoise.* In American practice viscosity is measured by the Saybolt Universal viscosimeter. This instrument gives the time in seconds required for 60 cc of an oil to flow through a given orifice at a certain temperature, usually at 130 or 210 F. The viscosity thus determined is called *specific viscosity* and designated t_s. In order to convert specific viscosity to absolute viscosity, another term is introduced, known as the *kinematic viscosity* Z_k, and measured in *centistokes.* The relation between kinematic and specific viscosities is given with a sufficient accuracy as

$$Z_k = 0.22t_s - 180/t_s \tag{26-3}$$

and the relation between absolute and kinematic viscosities is

$$Z = Z_k\gamma \tag{26-4}$$

where γ is the specific gravity of the oil relative to water.

Values of the specific gravity γ_{60} at 60 F compared with water at 60 F of several representative mineral oils used in internal-combustion engines are given in Table 26-1. In order to find

TABLE 26-1.—CHARACTERISTICS OF LUBRICATING OILS, FIG. 26-17

Designation	Oil specifications	SAE No.	Specific gravity at 60 F γ_{60}
A	Automobile oil, light	10	0.8894
B	Automobile oil, all-year	20	0.9036
C	Automobile oil	20	0.9254
D	Diesel oil	30	0.9250
E	Automobile oil, heavy	40	0.9275
F	Diesel oil	40	0.9285
G	Airplane oil 100	60	0.8927
H	Transmission oil	110	0.9328

the absolute viscosity of an oil by expression (26-4) at a certain temperature t F, the specific gravity γ_t of the oil at this temper-

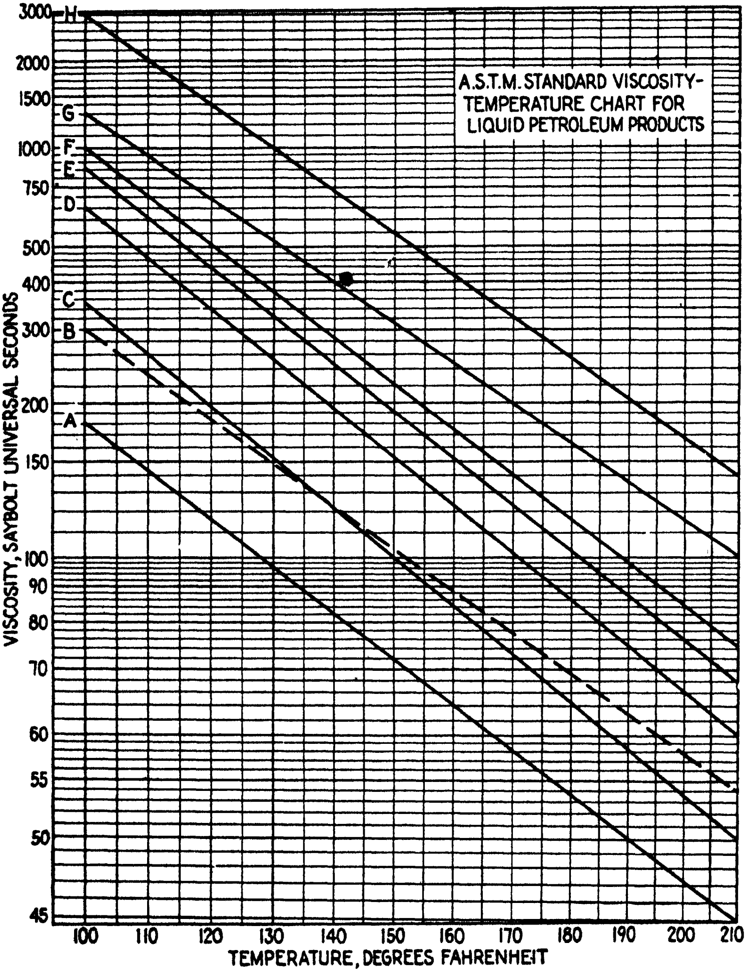

FIG. 26-17.—Viscosity-temperature chart.

ature must be found from the expression

$$\gamma_t = \gamma_{60} - 0.00035(t - 60) \qquad (26\text{-}5)$$

A special chart, Fig. 26-17, in which the viscosities are represented by straight lines, permits finding the viscosity of an oil at any temperature, if its viscosities at some two temperatures are known. The chart refers to the same typical oils for which data are given in Table 26-1.

Pressure.—In bearing calculations the term pressure is applied to the load F on a bearing divided by the projected journal area DL, where D is the diameter and L the length of the bearing,

$$p = F/DL \tag{26-6}$$

Experience shows that there are limits for p that should not be exceeded if perfect lubrication is to be maintained. The variation of the bearing pressure in internal-combustion engines complicates the determination of the safe pressure, which is found by means of polar diagrams. Data given in Table 26-2 may serve as a preliminary guide for the allowable pressures. The length-over-diameter ratio has a decided influence on the allowable pressure. The optimum ratio is $L/D = 0.75$, Fig. 26-18;[1] a decrease to $L/D = 0.5$ lowers the load factor k_l to about 0.9, because of a greater relative oil leakage at the ends; an increase to $L/D = 1.2$ again lowers k_l to about 0.9 because in a

TABLE 26-2.—DESIGN DATA FOR BEARINGS

Type of engine	Kind of bearing	Highest allowable pressure p psi	Suitable Z centipoise	Minimum Zn/p	Clearance c/D	Length to diameter ratio L/D
Automotive and aircraft engines	Main	800^{b} –1700^{c}	7	15	0.001	0.6–1.8
	Crankpin	$1500^{a,b}$–3500^{c}	to	10	0.001	0.6–1.4
	Piston pin	$2300^{a,b}$–5000^{c}	9	8	0.001	1.5–2.0
Gas and oil engines, four-stroke	Main	$700^{a,b}$–1200^{c}	15	20	0.001	0.6–2.0
	Crankpin	$1400^{a,b}$–2000^{c}	to	10	<0.001	0.6–1.5
	Piston pin	$1800^{a,b}$–2400^{c}	65	6	<0.001	1.5–2.0
Gas and oil engines, two-stroke	Main	$500^{a,b}$– 800^{c}	20	25	0.001	0.6–2.0
	Crankpin	$1000^{a,b}$–1500^{c}	to	12	<0.001	0.6–1.5
	Piston pin	$1200^{a,b}$–1800^{c}	65	8	<0.001	1.5–2.0

[a] Ring or drop oiler.

[b] Splash as scraper lubrication.

[c] Pressure lubrication.

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[1] *Mech. Eng.*, vol. 64, No. 6, p. 439, June, 1942.

longer bearing it becomes more difficult to obtain a uniform pressure p.

Bearing characteristic number is the name applied to the term Zn/p, where n is the journal speed, rpm. As shown later, Zn/p influences friction in the bearing and the oil-film thickness. Minimum values for Zn/p, as found from experience, are given in Table 26-2.

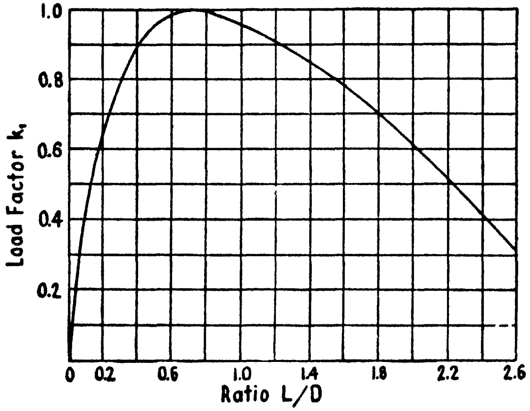

FIG. 26-18.—Effect of L/D ratio on load factor.

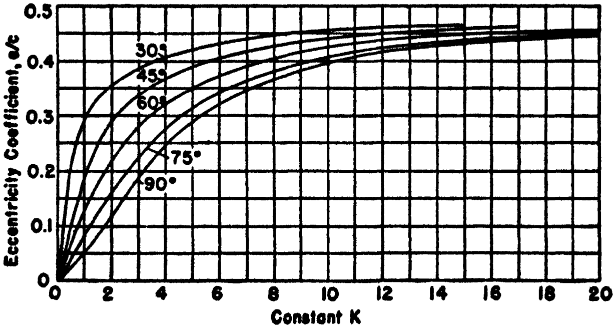

FIG. 26-19.—Chart for determination of the influence of eccentricity.

Oil-film Thickness.—The minimum oil-film thickness h_m, Fig. 26-14, of a bearing in operation can be expressed by the clearance c, which is the difference between the bore of the bearing and the shaft diameter, and the distance between the center of the shaft and that of the bearing, termed the *eccentricity* e.

Evidently $h_m = 0.5c - e$; this can be presented as

$$h_m = 0.5c(1 - 2e/c) \tag{26-7}$$

The magnitude of e is a function of pressure p, relative clearance c/D, oil viscosity Z, journal speed n, and the angle α, Fig. 26-14, between the edge m of the bearing relief, where the oil film starts, and the direction of the load. The ratio e/c, called the *eccentricity coefficient*, can be found from a chart, Fig. 26-19, in which the abscissas are proportional to a constant K. This constant can be computed from the expression[1]

$$K = 198(1000c/D)^2/(Zn/p) \tag{26-8}$$

The angles designating the curves in Fig. 26-19 refer to the angle α of Fig. 26-14. After K is computed, the ordinate of the corresponding curve gives directly the eccentricity coefficient e/c, and expression (26-7) permits finding the corresponding oil-film thickness h_m. The minimum thickness h_m that must be maintained for a perfect lubrication is around 0.0001 in. for finely bored small bronze bushings; at ordinary conditions, a steel journal in a babbitted shell, a thickness of at least 0.00075 in. should be maintained if n is high and not less than 0.0004 in. if n is low.[2]

Friction.—The expression for computing the coefficient of friction between a rotating journal and a bearing is[3]

$$f = K_a(Zn/p)(D/c)10^{-10} + \Delta f \tag{26-9}$$

The constant K_a may be set equal to 1.31β, where β is the circumferential length of the bearing in degrees. Thus for a full bearing, with two side relieves of 15° each, $\beta = 330°$, $k_a = 433$. For bearings having L/D from 0.75 to 2.8, $\Delta f = 0.002$.[4] Expression (26-9) is valid for $f \geqq 0.01$. If it gives $f < 0.01$, to be on the safe side $f = 0.01$ should be used.

Work of friction W_f is equal to the product of the frictional force and the distance through which it moves. The frictional force is the load on the bearing, LDp, times f. The distance through which the force travels in a unit of time is the peripheral velocity of the journal, $\frac{1}{12}\pi Dn$, fpm. Thus the work in foot-pounds per minute is

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[1] KARELITZ, G. R., *Trans. ASME*, 1930, APM-52-5, p. 59.

[2] *Trans. ASME*, 1930, APM-52-5, p. 58.

[3] MCKEE, S. A., and MCKEE, T. R., *Trans. ASME*, 1929, APM-51-15, p.164.

[4] MCKEE, S. A., *Mech. Eng.*, vol. 47, p. 1335, December, 1927.

$$W_f = \frac{1}{12}\pi fpLD^2n \tag{26-10}$$

Oil Temperature.—The work of friction is transformed into heat, which raises the temperature of the oil. This heat must be either dissipated to the surrounding air or, in the case of pressure-oil circulation, given up to the oil cooler. Practice shows that the temperature of a bearing should be kept, if possible, below 140 F and at any rate should not exceed 180 F. The usual temperature difference between oil leaving an engine and entering it is 10 to 20 F; the maximum temperature rise allowed in some heavy-duty high-speed engines is 40 F.

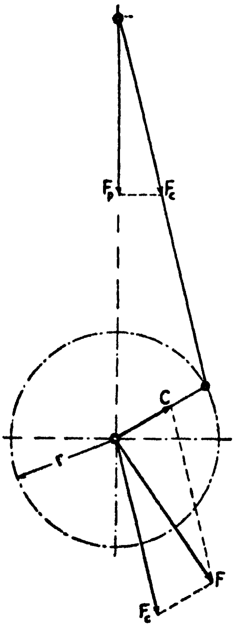

FIG. 26-20.—Forces acting on a main bearing.

26-5. Bearing Loads.—It should be noted that there are three types of bearings in an engine whose design requires particular attention, namely, those supporting: (*a*) main journals, (*b*) crankpins, and (*c*) piston pins.

All these bearings are subjected to a combination of gas pressures and forces of inertia; however, there is some difference in their application.

Main Bearings.—The load F acting on a main bearing is found as a resultant of the components F_c of the net effort F_p on the piston along the axis of the connecting rod and of the centrifugal force C, Fig. 26-20. The centrifugal force may be determined by equation (25-2), where W is the weight of the big end of the connecting rod together with the unbalanced part of the crank, pounds, and r is the crank radius, feet. If, as in most compression-ignition engines, each crank is located halfway between two main bearings, it can be assumed that each bearing receives one-half of the resultant load. However, each bearing, except the end bearings, has a one-half load coming from each of the cranks between which the bearing is located. These loads must be added geometrically, as their direction is different. If more than one crank is located between the bearings, the loads from each crank first must be properly distributed between the two adjacent bearings, taking into account the distances from the center of each crank to the centers of the two bearings, and then combined with the loads produced by the cranks located on the other side of the two bearings under consideration. The simplest procedure for this is the force-polygon method. As in all crankshaft computations, it is assumed that the action of any force acting at any point of a crankshaft does not extend beyond the two adjacent bearings. The resultant forces computed for every 10 to 30 crank-angle degrees for the complete cycle are plotted as a polar diagram The mean and maximum loads are found then.

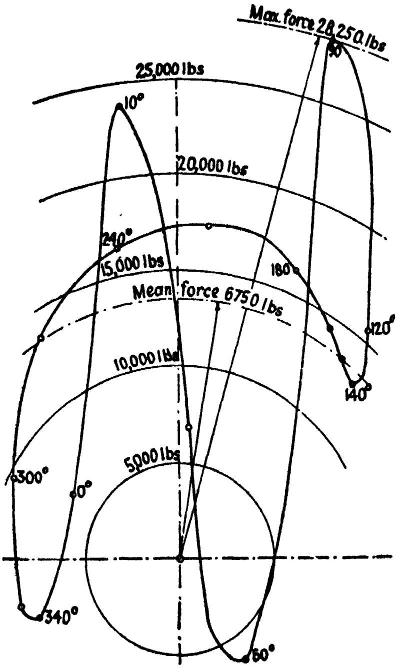

FIG. 26-21.—Polar diagram of forces acting on a main bearing of a two-stroke oil engine.

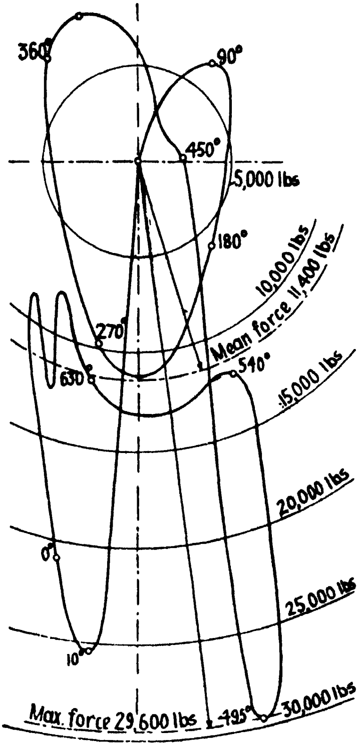

FIG. 26-22.—Polar diagram of forces acting on a main bearing of a four-stroke oil engine.

The shape of the polar diagrams thus constructed can vary not only from engine type to engine type but, in a multicylinder engine, from one main bearing to the next one, and it is impossible to speak of typical shapes. Only two samples will be shown. Figure 26-21 is a diagram for a two-stroke compression-ignition oil-engine main bearing, with the adjacent cranks at 90°, and Fig. 26-22 is a diagram for a four-stroke oil-engine main bearing.

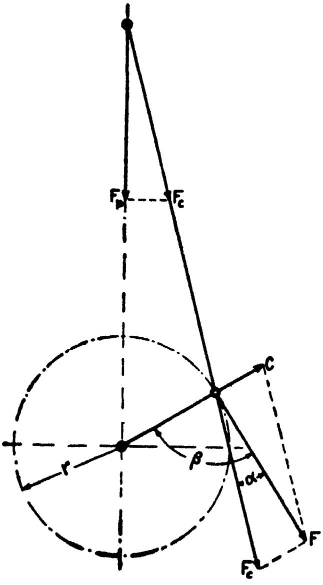

FIG. 26-23.—Forces acting on a crankpin bearing.

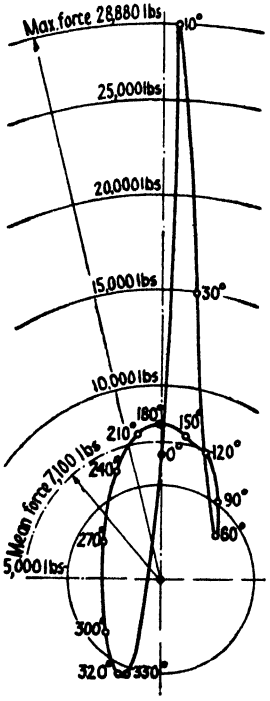

FIG. 26-24.—Polar diagram of forces acting on a crankpin bearing.

Crankpin Bearing.—The load F acting on the crankpin bearing, which is a part of the big end of the connecting rod, is found in a way similar to the one on the main bearing as the resultant of the component F_c of the net effort on the piston F_p, along the axis of the connecting rod, and of the centrifugal force C of the big end of the connecting rod, Fig. 26-23.

The centrifugal force may be determined by equation (25-2), where W is the weight of the big end of the connecting rod, pounds, and r the crank radius, feet.

In plotting a polar diagram of the forces F the angles α, Fig. 26-23, must be used. Figure 26-24 shows a typical diagram for a two-stroke compression-ignition oil engine. It should be noted that, if bending stresses or wear of the crankpin itself is investigated, the polar diagram will be different, because the angles β, Fig. 26-23, instead of angles α, must be used.

If two or more connecting rods are joined to the same crankpin, the resultant of the net efforts in all cylinders must be found for each crankpin position and this resultant combined with the centrifugal force of the big ends of all connecting rods. The resultant of all net efforts may be found by a force polygon.

Piston-pin Bearing.—The load acting on the piston pin is a component, along the axis of the connecting rod, of the algebraic sum of the gas pressure and of the inertia force of the piston assembly, without the connecting rod. This force may be assumed to be perpendicular to the piston-pin axis and oscillates through an angle equal to twice the maximum connecting-rod angularity angle. Figure 26-25 gives a typical polar diagram of the piston-pin loading of a two-stroke engine.

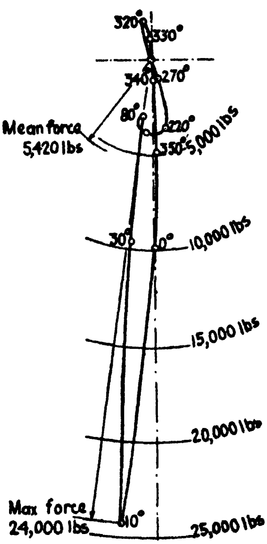

FIG. 26-25.—Polar diagram of forces acting on a wristpin.

26-6. Bearing Design.—The main problem in designing a bearing consists in finding the minimum diameter and length which will give satisfactory operation and small, if any, wear for the given operating conditions, maximum and mean load, rotative speed, and a certain lubricating oil. If the latter is not given, it must be selected on the basis of viscosity as given in Table 26-1.

The consecutive design steps are:

1. *Lubrication.*—The type of lubrication to be used must be selected:

(*a*) ring oiling or drop oilers,
(*b*) splash lubrication, pure or assisted by gravity,
(*c*) force-feed lubrication,
(*d*) pressure lubrication.

2. *Pressure.*—Depending upon the type of lubrication selected and the type of engine, the maximum permissible pressure p is selected, Table 26-2.

3. *Main Dimensions.*—With p selected and maximum continuous load F given, the necessary projected area is found

$$A = DL \geqq F/p \tag{26-11}$$

If the diameter D was determined from other conditions, equation (26-11) gives the required length L of the bearing and the ratio L/D is checked by Table 26-2. Otherwise, the ratio L/D may be selected from Table 26-2, and, with A computed by expression (26-11), the values for D and L are found. The influence of the ratio L/D upon p should be taken into account, using Fig. 26-18. For the main crankpin and piston-pin bearings the loads F must be found from polar diagrams, as explained in Sec. 26-5. For high-speed engines the mean force as indicated in Figs. 26-21, 26-22, 26-24, and 26-25 can be used as the design load F, because the duration of action of the maximum force F_{max} is so short that there is not enough time for the oil-film thickness to be materially reduced. However, with a lowering of the engine speed, rpm, the influence of F_{max} increases. For engines operating under 1200 rpm this influence may be taken into account by increasing the design load F over the mean force F_{mean}, found from the polar diagram, by an additional value ΔF, found from the expression

$$\Delta F = (F_{max} - F_{mean}) \frac{1200 - n}{1200} \times \frac{\gamma}{180c} \tag{26-12}$$

where n is the engine speed, rpm,

γ is the crank angle on the polar diagram during which the load exceeds F_{mean}, *i.e.*, changes from F_{mean} to F_{max} and back to F_{mean},

c is the number of strokes in the engine cycle, two for a two-stroke engine and four for a four-stroke one.

4. *Temperatures.*—The temperature t_1 of the oil entering the bearings and t_2 leaving them are assumed within the limits stated in Sec. 26-4. From the specific viscosity t_s of the selected oil and its gravity γ_{60}, the absolute viscosity Z at the temperature t_2 is computed and the bearing characteristic numbers Zn/p are found for the mean pressure p. The values of Z and Zn/p are

checked by Table 26-2, and if necessary the assumption of t_1 and t_2 are changed.

5. *Bearing clearance* c is selected, Table 26-2, and the oil-film thickness h_m is computed from equation (26-8), Fig. 26-19, and expression (26-7). It is safer to use p_{max} in these calculations, but if a small wear is considered permissible, the mean pressure p may be used. If h_m as found is not large enough, one or several of the factors, t_2, Z, p, or angle α, must be changed and h_m recalculated until it becomes large enough for a safe bearing operation, as stated in Sec. 26-4.

6. *Friction.*—The coefficient of friction f is computed by equation (26-9), and the work of friction W_f by equation (26-10). From the oil-temperature rise $t_2 - t_1$ and friction work W_f, assuming a specific heat of oil $c = 0.5$ Btu/lb-°F and knowing its gravity γ, the amount of oil Q, gpm, which must be circulated, is computed. This amount Q should be checked with the limits as found in practice and given in Sec. 26-3.

If ring-oil bearings or splash lubrication is used, the amount of heat corresponding to W_f must be dissipated by the outside surface of the bearing. The temperature differences between the oil and the bearing surface, and between this surface and the air, must be established; the film coefficient of heat transfer from the metal surface to the air must be found, and from these data the amount of heat, Q_d, dissipated can be found. Evidently Q_d must be equal to W_f. If $W_f > Q_d$, the temperature t_2 of the oil will go up, increasing Q_d and lowering W_f, because of a decrease of Z and f, until $W_f = Q_d$. The oil-film thickness h_m must be recalculated for the new temperature t_2'. If h_m is found to be too small, the projected area LD must be increased in order to lower the pressure p, increase the number Zn/p, and increase the area of the heat-dissipating surface. All calculations must be repeated until a satisfactory value of h_m is obtained.

26-7. Other Parts. *Camshaft.*—Depending upon the type of engine, and other considerations, the camshaft bearings are lubricated by all conceivable methods: sight-feed oilers, ring oilers, automatic grease cups; when located inside an enclosed crankcase—by splash or feed lines from the main oil-pressure system. In large horizontal engines the long camshafts combined with lay shafts run in ball bearings that are grease packed and require the renewal of grease only once in several months.

Cams.—On account of the blows that a cam, especially the exhaust cam, receives at each cycle great care in regard to lubrication is necessary to prevent rapid wear of the cam and its pin. In low-speed engines the cams often simply dip in an oil bath, the level in which is kept constant by a branch of the pressure oil pipe and an overflow. In engines with an enclosed crankcase, the oil splash and mist keep the cams and their bushings lubricated.

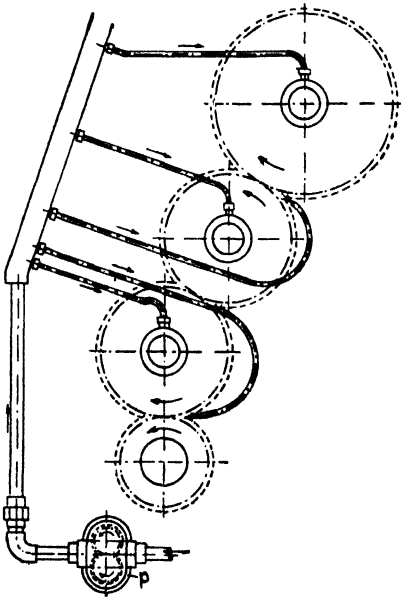

Fig. 26-26.—Lubrication of a gear drive.

Camshaft Drive.—To decrease noise and wear, the drive gears must either run in an oil bath or receive a small stream of oil from the pressure system. In Fig. 26-26, all bearings have pressure lubrication, and small streams of oil taken from the main pressure system are directed on the teeth of mating gears. In enclosed crankcases this stream can be obtained from splash oil guided by special catchers. The same schemes are used for governor drives and silent chain drives.

Valves.—The guides and stems of valves must be lubricated only very sparingly, especially those of the exhaust valve. A couple of drops per minute or a grease cup with spring pressure is all that is necessary.

26-8. Problems.—**1.** Determine the amount of cylinder-lubricating oil in gallons per hour for a four-stroke two-cylinder horizontal double-acting gas engine of 650 hp per cylinder and the size of each copper tube sufficient to feed the lubricating points.

2. Determine the amount of circulating oil for a four-stroke vertical four-cylinder 800-hp oil engine. Find the size of the plunger pressure pump if its stroke-bore ratio is 1 and the scavenge pump has the same stroke. The engine runs at 250 rpm and both pumps are driven from the camshaft at its speed.

3. Determine: (*a*) the amount of circulating oil for a two-stroke four-cylinder 180-hp oil engine running at 1200 rpm; (*b*) the size of the gear pumps, pressure, and sump scavenge, if the pumps are driven by step-up gears with a 1:1.25 ratio.

4. Find the size of the pressure-oil pipe and scavenge pipe for the engine in problem 3.

5. Determine the amount of water necessary for the oil cooler for the engine in problem 2 if the water is admitted with a temperature of 70 F and discharged with a temperature of 80 F. Also find the necessary cooling area of the oil cooler, using data from Secs. 12-5 and 21-2.

6. Determine the size of the gear pumps, pressure and sump scavenge, for a 200-hp radial air-cooled airplane engine. Engine speed 1700 rpm; pump runs with a reduction of 2:1.

7. Find the sizes of the pressure oil pipe and suction (scavenge) pipe for the engine in problem 6.

8. Determine the size of the gear pumps, pressure, and pump scavenge, for a four-stroke motorboat compression-ignition oil engine of 240 hp at 1200 rpm; the pumps run with a speed step-up of 1:1.3.

9. Find the oil pressure in the wristpin bearing due to the centrifugal force in a banjo lubricator of a horizontal engine 6½-in. bore × 10½-in. stroke, 550 rpm; the length of the connecting rod is 4.5 times the crank radius.

10. Find the engine speed when the centrifugal force would become insufficient to raise the oil to the wristpin in the engine in problem 9.

11. Find the oil pressure in the wristpin bearing, produced by the centrifugal force and force of inertia, of a vertical 8-in. × 10-in. engine running at 900 rpm; the oil pressure in the system is 60 psi; the length of the connecting rod from center to center is 22½ in.

12. Find the limits of pressure fluctuation in the wristpin bearing of the engine of problem 11 during one revolution; the oil passage drilled in the connecting rod communicates with the hole drilled in the crankshaft pin 15° before top dead center.

13. Plot a polar diagram of loads on the piston-pin bearing of an 8-in. × 12-in. gas engine operating at 240 rpm; the piston weighs 100 lb.; as indicator diagram use Fig. 13-4.

14. Plot a polar diagram of loads on the crankpin bearing of the engine of problem 13; the weight of the big end of the connecting rod is 60 lb.

CHAPTER 27

RUNNING GEAR

27-1. Trunk Pistons.—*The functions* of a trunk piston are:

1. To transmit the gas pressure to the crankshaft.
2. To take the side pressures due to angularity of the connecting rod.
3. To seal the inside of the cylinder from the crankcase.
4. To dissipate heat absorbed by the piston top during combustion and early part of expansion stroke.

Design Objects.—In designing a piston to meet these requirements the following objects must be sought:

1. Strength of the piston, particularly of its head.
2. Sufficient projected side area, and rigidity of the barrel.
3. Minimum work of friction.
4. Sealing of the working space against escape of gases.
5. Preventing the entrance of lubricating oil into the combustion space.
6. Good dissipation of the heat to the cylinder walls.
7. Minimum weight.

The design objects are listed not in the order of their importance but to conform with the order of functions as listed above. Good heat dissipation is one of the most important design requirements.

Materials used to make trunk pistons are in the order of their importance: cast iron, cast aluminum, forged aluminum, cast steel, and forged steel.

Cast iron is an excellent material; its main drawback is that it gives a slightly heavier piston than aluminum. However, with a proper design the difference is only about 10 to 20 per cent. Cast-iron pistons produce less cylinder-liner wear than aluminum ones,[1] especially if they are tin-plated.

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[1] *Automotive Ind.*, Jan. 16, 1937, p. 81; *Z. Ver. deut. Ing.*, vol. 81, p. 610, 1937.

Cast aluminum alloy gives better heat dissipation and lighter weight but costs considerably more than cast iron. The strength is about the same as that of cast iron.

Forged-aluminum pistons are stronger and still lighter. They are used for aircraft engines and heavy-duty high-speed compression-ignition oil engines.

Alloy cast-steel pistons are used in some automotive engines and require liners of great surface hardness. The same is true of forged-steel pistons used in some aircraft engines.

Piston Head.—The thickness t_1 of the head or crown can be computed, considering it a flat round plate of uniform thickness fixed at the edges, from the formula

$$t_1 = 0.43D \sqrt{p/S} \qquad (27\text{-}1)$$

where D is the cylinder bore, in.,
p is the maximum pressure during combustion, psi,
S is the allowable stress in bending, psi.

A stress of 5500 psi can be allowed when using a good close-grained cast iron or an aluminum alloy with an ultimate tensile strength of 20,000 psi. If the material used has an ultimate strength of 30,000 psi, such as nickel cast iron, semisteel, or special aluminum alloy, normalized, then S can be taken as 8000 psi. For forged-steel heads S can be raised to 12,000 psi. The piston head has often four or six radial ribs a, Fig. 27-1, of a thickness t_2, one-third to one-half the thickness t_1 of the head, but it is safer not to consider these ribs when computing t_1.

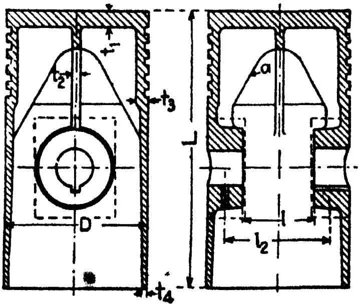

FIG. 27-1.—Trunk piston of a gas engine.

TABLE 27-1.—THICKNESS OF PISTON HEAD

Type of engine	Piston material	Four-stroke	Two-stroke
Compression-ignition oil engines...	Cast iron	$0.11D$–$0.13D$	$0.16D$–$0.18D$
Compression-ignition oil engines...	Aluminum	$0.13D$–$0.16D$	$0.17D$–$0.20D$
Spark-ignition gas engines........	Cast iron	$0.12D$–$0.14D$	$0.20D$–$0.23D$

In order to provide a sufficient cross area for the heat flow, the head often is made thicker than required by formula (27-1). The thickness of the head as found in practice in various engines is given in Table 27-1.[1] The thickness of water or oil-cooled pistons is made smaller, in accordance with formula (27-1).

For cast-iron automobile-engine pistons an empirical formula may be used

$$t_1 = 0.12 + 0.03(D - 2) \tag{27-2}$$

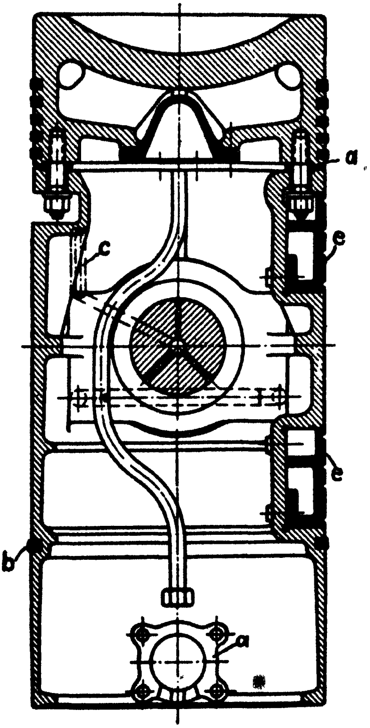

FIG. 27-2.—Oil-cooled oil-engine piston. Scale 1:13.

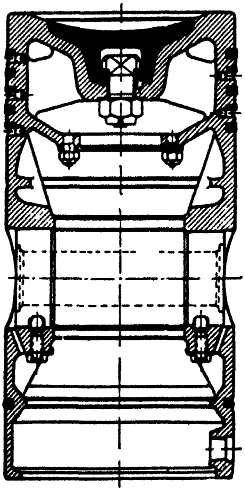

FIG. 27-3.—Oil-engine piston with mushroom center insert. Scale 1:19.

In large oil engines the upper part of the piston is sometimes made as a separate piece fastened by studs *a* to the lower part, Figs. 27-2 and 27-7. The object of such a construction is to use for the head a material with a higher strength, such as cast steel.

In cast-iron pistons with a steel head the diameter of the head is made so that it will not touch the cylinder walls, even at the highest temperature. Another advantage of this construction is the moderate cost of exchanging it when the top of the piston develops cracks, which may happen because of the high temperature stresses, or is burned by the fuel oil hitting it.

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[1] BOYER, R. L., and KUIVINEN, T. O., "Stress and Deflection in Reciprocating Ports of Internal Combustion Engines." Paper presented at the ASME Conference of the Oil and Gas Power Division, June 19–22, 1940.

Figure 27-3 shows an oil-engine piston with a so-called *mushroom* steel insert for the same purpose, and Fig. 27-4 shows an aluminum piston with a cast-iron top.

Piston Barrel.—The maximum thickness t_3 of the barrel, Fig. 27-1, may be determined by the formula

$$t_3 = 0.18 + 0.03D + b \tag{27-3}$$

where b is the depth of the ring grooves, about $\frac{1}{64}$ in. greater than the thickness of the rings. The wall thickness should decrease toward the open end of the piston down to

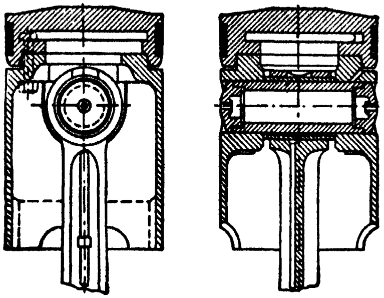

FIG. 27-4.—Beardmore composite piston. Scale 1:10.

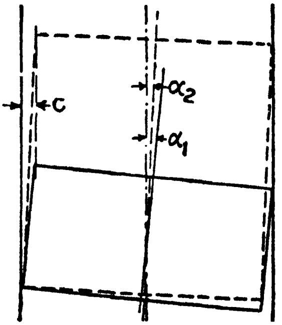

FIG. 27-5.—Influence of piston length.

$$t_4 = 0.25t_3 \text{ to } 0.35t_3 \tag{27-4}$$

in order to make the piston lighter.

The length L of the barrel may be determined from the condition that the side pressure resulting from the angularity of the connecting rod, for low-speed engines, should not exceed 30 to 40 psi, not counting the upper part of the piston with the rings. The maximum side pressure should be found from an actual or theoretical indicator diagram. It occurs at a crank angle of about 20 to 25°. The normal pressure varies from 0.03 F_{max} to 0.1 F_{max}. A longer piston runs more quietly, has less slap, as the increase of its length decreases the angle of lateral movement from α_1 to α_2, Fig. 27-5, with the same clearance c between piston and cylinder necessary to take care of expansion. Figure 27-7 shows such a piston, which is so long that it requires an extra long connecting rod, $l/r = 5.25$, to clear the piston skirt.

Another means of decreasing the lateral movement and thus avoiding piston slap is the use of separate adjustable glide pieces, *e, e,* Fig. 27-2, applied only in very large engines.

To reduce the weight of the reciprocating parts and of the whole engine, pistons of high-speed engines are made considerably shorter. This results in side pressures up to 70 psi and slightly greater wear.

Piston Pin.—To obtain uniform distribution of pressure between piston and cylinder, the piston pin, often called *wrist-pin,* should be placed in the center of the piston part carrying the load. This would be halfway between the edge of the last ring groove and the end of the piston. However, to shorten the height of the engine, the piston pin is usually moved nearer to the top, Fig. 27-3, sometimes about halfway between the two ends, Fig. 27-2, and sometimes even nearer to the top end, Fig. 27-4. In vertical engines placing the piston pin nearer to the top gives a quieter running engine.

The length l, Fig. 27-2, of the piston pin is restricted by the cylinder bore, its diameter d is determined by the allowable bearing pressure p_b, from the relation

$$ldp_b = 0.7854D^2p_{max} \qquad (27\text{-}5)$$

The space in a piston is so limited that in the conventional design a high pressure must be permitted, up to $p_b = 1800$ psi for gas engines and 2200 psi for oil engines, as shown in Table 26-2.

The bending stress is determined considering the pin as a beam uniformly loaded for the distance l and supported at the centers of the bosses, making the length of the beam between supports l_2, Fig. 27-1. For the sake of simplicity and to be on the safe side, l_2 can be expressed as

$$l_2 = l + (D - l)/2 = (D + l)/2 \qquad (27\text{-}6)$$

and the bending moment, under these conditions, at the center of the pin

$$M = F(l_2/2 - l/4)/2$$

Substituting expression (27-6) for l_2 gives

$$M = FD/8 \tag{27-7}$$

and the stress

$$s = M/Z \tag{27-8}$$

In order to reduce the weight the piston pin is made hollow and the section modulus is then

$$Z = \pi(d^4 - d_0^4)/32d \tag{27-9}$$

The stress s should not exceed 12,000 psi for case-hardened carbon steel and 20,000 psi for heat-treated alloy steel.

The high pressures necessitate the use of bushings of special high-grade bronze, and even these show fast wear, as do the stationary bearings in the piston bosses.

In high-speed oil engines the piston pins usually are of the full-floating type, with bronze bushings in the connecting rod and in the piston bosses and either with bronze or aluminum end plugs or spring retainers to prevent the pin from touching and scoring the cylinder liner.

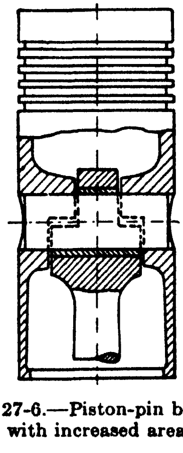

FIG. 27-6.—Piston-pin bearing with increased area.

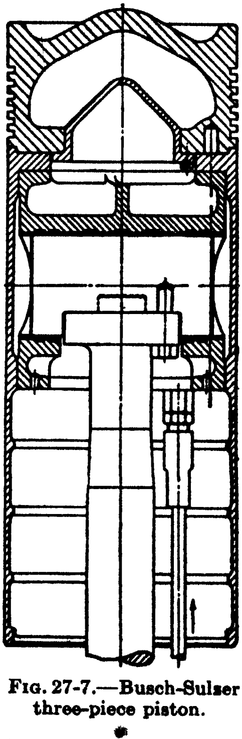

FIG. 27-7.—Busch-Sulzer three-piece piston.

In trunk pistons in which the pressures act in one direction, as in two-stroke engines or in low-speed engines with light pistons, the bearing pressures can be lowered using the construction shown in Fig. 27-6. A further step in this respect is the piston shown in Fig. 27-7, made of three pieces, the head, barrel, and a center piece to take the piston pin's pressure. The head is a steel forging; the wristpin is bolted to the connecting rod and increases the bearing area so as to lower the pressure down to about 1250 psi. This pressure also permits the use of a babbitt bearing surface.

In this construction the working pressures are transmitted directly to the connecting rod, and the barrel is neither strained nor distorted by wristpin bosses.

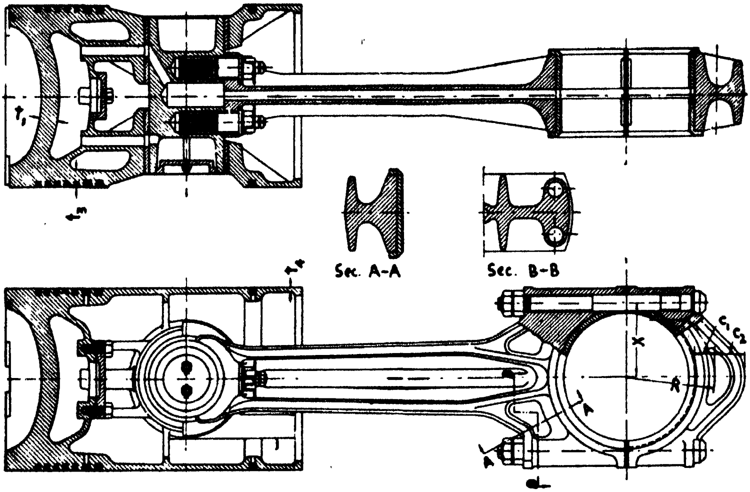

FIG. 27-8.—Piston and connecting rod of a Cooper-Bessemer oil engine.

A similar design for a four-stroke medium-speed engine is shown in Fig. 27-8. The piston pin is bolted to the connecting rod and works in a bushing pressed into the piston. The piston head is cooled by oil, admitted through the drilled connecting rod and discharged into the crankcase.

In the conventional design the barrel should be relieved around the piston pin as shown by dotted lines in Fig. 27-1 and still more clearly in Fig. 27-9.

The methods of fitting the piston pin to the piston and holding it in place can be seen in the different drawings.

In automotive and aircraft engines the piston pin usually is clamped in the connecting rod and sometimes is made floating both in the piston bosses and in a bushing that is inserted into the end of the connecting rod.

For larger engines the piston pin is made of mild steel, casehardened, and ground to final size; for small high-speed engines it is made of nickel or vanadium steel, hardened and ground.

Lubrication of the piston pin is provided either by forcing oil under pressure from the hollow crankshaft and up the hollow connecting rod, or by collecting the oil from the cylinder surface and leading it to the hollow wristpin and from there to the bushing, Fig. 27-2.

Design of the piston pin should be based on data given in Table 26-2 and on the polar diagram of loads.

Friction.—The friction between piston and cylinder is reduced by using proper materials, smooth finish of the surfaces, piston rings with reasonably free pistons, minimum pressure, good lubrication, and reasonably high temperatures of the oil.

The best materials for pistons are close-grained cast iron and special aluminum alloys. Smooth surfaces are obtained in the cylinders by reaming or, better, by honing. Grinding of the cylinders is less satisfactory since small particles of grinding material lodge in the cylinder surface and increase friction and wear of the pistons. The pistons usually are ground. The pressure necessary to seal the wearing surface is small, about 5 to 6 psi in low-speed engines and decreasing down to 3.5 psi in high-speed engines. Gas pressures penetrating behind the rings increase the wall pressure and friction losses considerably. This is the reason why many engine builders prefer simple snap rings to special composite rings in which a greater sealing capacity is obtained through gas pressure.

Because the side pressure of the pistons is small, there is no danger of destruction of the oil film if its temperature does not exceed 190 to 200 F. On the other hand, the viscosity of lubricating oils increases very rapidly with a decrease of its temperature.

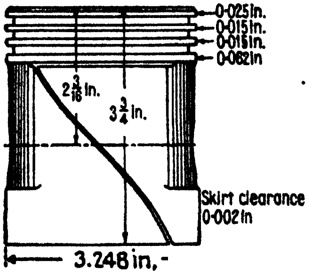

FIG. 27-9.—Split-skirt piston.

To take care of the large expansion coefficient of aluminum and at the same time to avoid loose pistons at low temperatures, aluminum-alloy pistons often are made with split skirts, Fig. 27-9, sometimes with steel rings cast into the aluminum to give rigidity and to force the aluminum to expand as though it were steel.

Piston Rings.—Simple one-piece snap rings with a wall pressure p_w between 3.5 to 6 psi give the best results. There is no strict rule as to their number—automobile and aircraft engine pistons have three rings, sometimes four; stationary compression-ignition engines have five to seven piston rings. The radial width of a cast-iron snap ring can be found from expression

$$b_r = D\sqrt{3p_w/S_b} \qquad (27\text{-}10)$$

where the allowable stress S_b for the cast iron is taken very high, from 12,000 to 16,000 psi, in order to avoid an excessive stress when slipping the ring into the groove.

The thickness h of the rings can be taken from $0.7b_r$ to b_r or based upon the number of rings i used,

$$h = D/10i \qquad (27\text{-}11)$$

In general thinner piston rings have the following advantages:[1]

(*a*) less wear of the lands;

(*b*) better sealing action;

(*c*) decreased friction, especially at high speeds;

(*d*) less piston length occupied by the rings.

The first ring groove from the top should be cut at a distance equal to t_1 or $1.2t_1$, Fig. 27-1. The lands between the ring grooves are made equal or slightly less than the ring thickness h.

The gap c between the free ring ends is made

$$c = 3.5b_r \text{ to } 4b_r \qquad (27\text{-}12)$$

The gap when the ring is in the cylinder should be $0.002D$ to $0.004D$.

In horizontal engines the rings must be located by pins to prevent them from turning, or all gaps will come to the top of the piston.

Rings suitable for any piston size and requirement can be procured from a number of manufacturers specializing in this product.

Oil pumping is prevented by rounding off the upper edges of the piston rings, Fig. 27-10*a*. The last ring is made as an oil scraper with a slight bevel, Fig. 27-10*b*, beveling off the lower groove edge and drilling small holes toward the inside of the piston to assist the removal of excess oil. In the case of excessive lubricating-oil consumption a more effective scraper ring with an undercut may be used, Fig. 27-10*c*. Piston rings with a high specific pressure act in a way similar to scraper rings, but their drawback is an increased cylinder wear. High-speed engines use oil rings with a circumferential slot for draining oil.

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[1] Diesel Piston Design Problems, *SAE Trans.*, December, 1942, p. 537.

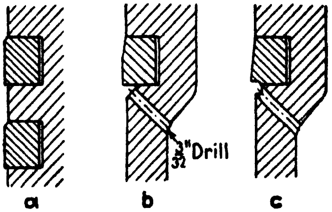


FIG. 27-10.—Methods of preventing oil pumping.

Heat Dissipation.—Proper heat dissipation is obtained by a sufficiently thick top and upper part of the barrel.

Light weight is obtained by using light alloys, although properly designed cast-iron pistons can be made almost as light and give better service at moderate speeds. The so-called *cut-away piston* and its modification, the *slipper piston*, Fig. 27-11, have not proved satisfactory on account of very high lubricating-oil consumption. Figure 27-11 shows the method of decreasing the side pressure by offsetting the wristpin by an amount *e*. However, this method is not used in modern engines.

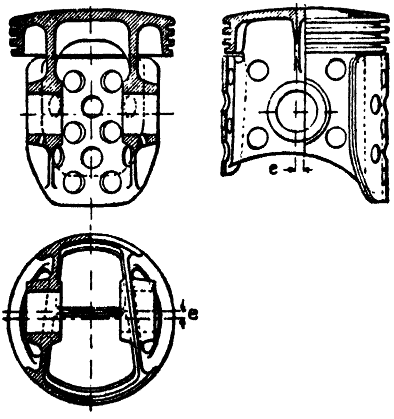


FIG. 27-11.—Slipper piston.

Of later years the designing and making of pistons for internal-combustion engines have become, to a large extent, a separate industry.

27-2. Pistons with Crossheads.—These pistons are used only in the larger and largest single- and double-acting engines and in most cases are water or oil cooled. Their construction looks more complicated, but their correct designing is easier than that of trunk pistons because their functions are reduced to transmitting the gas pressure to the crosshead and to sealing the working space. Most remarks made in regard to trunk pistons can be applied to crosshead or barrel pistons.

Single-acting Pistons.—The thickness of the crown may be figured from expression (27-1) and that of the barrel from expression (27-3). In four-stroke engines the barrel is made short, only to hold the six to eight compression piston rings, Fig. 27-17. In two-stroke engines a long skirt is added, Fig. 27-12, to act as a slide valve for the exhaust and scavenge ports. The lower part of Fig. 27-12 is a cross section by a horizontal plane of the frame, crosshead guide, and part of the crosshead shoe to show the cooling of the crosshead wristpin and piston. To show the water flow more clearly, the intake and discharge connections *a* and *b* are shown looking down, whereas actually they swing about a middle horizontal position.

Double-acting Pistons.—An example of a piston constructed for a horizontal four-stroke engine is shown in Fig. 21-10. A piston for a two-stroke engine is shown in Fig. 27-13.

Piston Rods.—The piston rod usually is made hollow, even if it does not serve for conveying the liquid for cooling the piston. The size of the rod is determined considering it a short column and using an expression derived from Ritter's formula

$$A = F[1 + (l/k)^2 S_e/(\pi^2 nE)]/S_d \tag{27-13}$$

where A is the section area of the rod, sq in.,
F is the maximum load upon the piston, lb,
n is the coefficient of end conditions,
l is the length of the rod, in.,
k is the radius of gyration of the rod section, in.,
S_e is the elastic limit or yield point of the material, psi,
E is the modulus of elasticity of the material, psi,
S_d is the allowable stress in compression, psi.

The coefficient n depends upon the method of fastening the ends: If both ends are fixed, $n = 4$; if one end is fixed and the

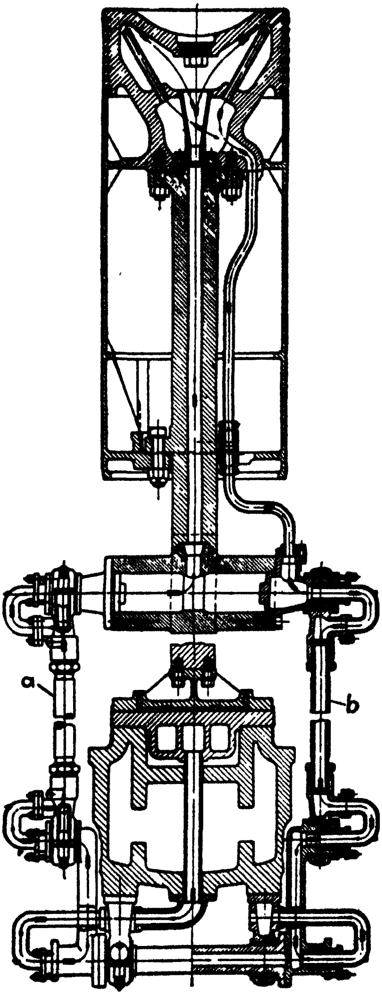


FIG. 27-12.—Piston and wristpin cooling of a two-stroke engine.

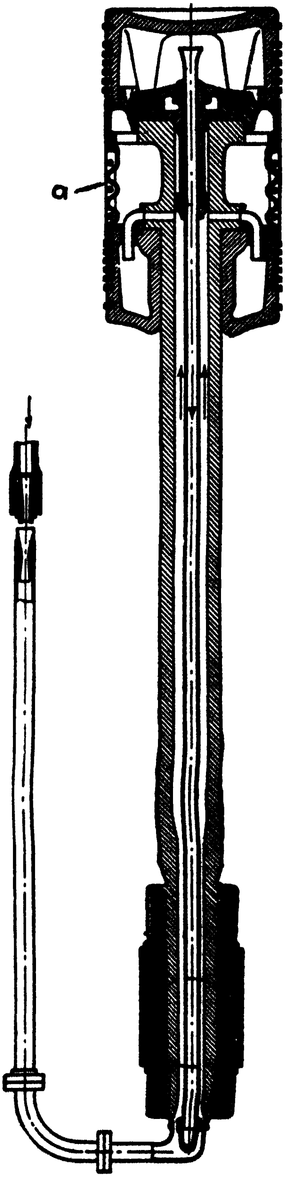


FIG. 27-13.—Piston and rod of a double-acting, two-stroke Sulzer oil engine.

other hinged, $n = 2$. The radius of gyration for a tube with an outside diameter d_1 and inside diameter d_2 is

$$k = 0.25\sqrt{d_1^2 + d_2^2} \tag{27-14}$$

The design stress S_d should not exceed 8000 psi for good-grade open-hearth steel and 12,000 psi for nickel or vanadium steel.

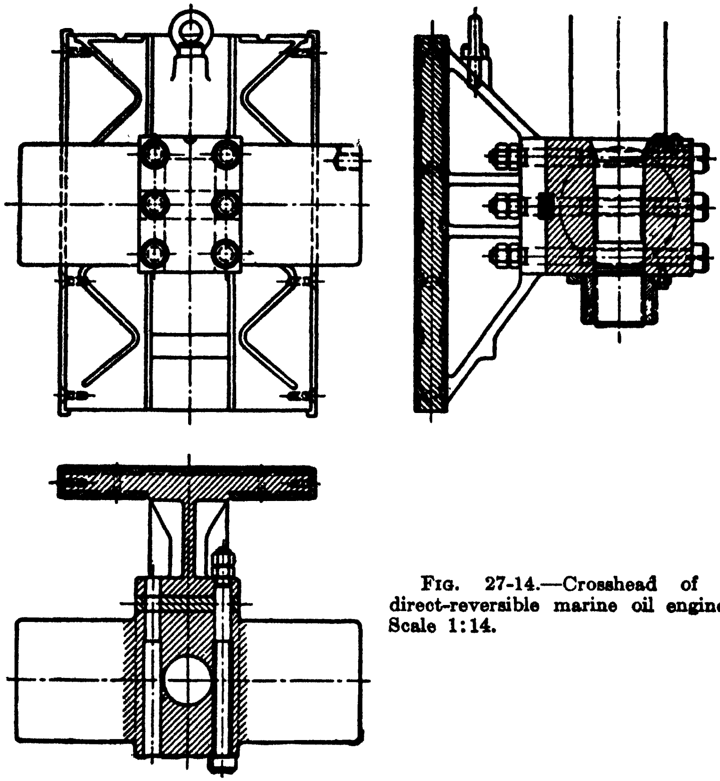


Fig. 27-14.—Crosshead of a direct-reversible marine oil engine. Scale 1:14.

The corresponding high safety factor also ensures the necessary rigidity.

Crosshead.—The side thrust coming on a crosshead may be determined in a way similar to the one in a trunk piston. Because better lubrication can be provided for a crosshead, and it operates practically at room temperature, considerably higher bearing pressures may be used, $p_c = 50$ to 75 psi, even up to 100 psi. The bearing surface is made either flat, Fig. 27-12, or flat with two surfaces, Fig. 27-14, for a reversible marine oil engine, or cylindrical as in most horizontal engines. The crosshead is babbitted; the guides are cast iron. The wear of the crosshead is taken up in some constructions by adjusting wedges but in most cases simply by inserting shims.

Stuffing Box.—The simplest and best type of stuffing box is a combination of metallic split rings held together by light spiral springs. In the present stuffing-box construction the troubles of early-day large horizontal engines are overcome, and, furthermore, stuffing boxes can be procured from manufacturers specializing in metallic stuffing boxes for compressors and internal-combustion engines.

27-3. Temperature Influence.—Three kinds of heat flow must be considered when designing an engine piston:

(*a*) absorption of heat from the gases,

(*b*) heat flow through the piston, and

(*c*) heat emission, chiefly to the cylinder walls.

Evidently, it is desirable to absorb as little heat as possible. The flat-top piston gives the least exposed surface to the combustion gases, thereby absorbing less heat and giving lower piston temperatures.

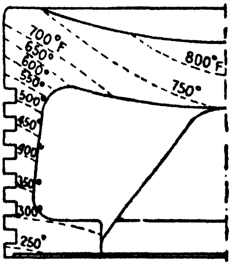


FIG. 27-15.—Temperatures in an oil-engine piston crown.

In order to obtain a good heat flow through the cylinder, the piston crown should be of uniform thickness, and the barrel, where it joins the crown, should have the same thickness. Built-up constructions give more resistance to the heat flow and therefore should be avoided.

In order to obtain good heat emission, the rings must be made narrow but deep and must be well-fitted; the barrel behind the ring grooves must have the thickness necessary for the heat flow.

Uncooled Pistons.—The maximum crown temperature even in the uncooled pistons is not high compared with that of the gases during the working and exhaust strokes. The heat transfer from the piston to the fresh charge is small, as shown in a previous discussion, constituting in four-stroke engines from 10 to 20 per cent and in two-stroke engines from 14 to 27 per cent of the total jacket heat. However, the piston temperatures are high enough to create dangerous temperature stresses unless special provisions are made in the design.

Figure 27-15 shows the temperatures in various points of the crown of a two-piece cast-iron piston. Figure 27-16 shows the mean temperatures in the walls of an aluminum trunk piston, curve *a* in the piston crown and *b* in the trunk part.

In a cast-iron piston the temperatures run about 200 to 300° higher at the crown center, about 100 to 150° higher at the edge. In the skirt they gradually taper down to about 200 F.

The temperatures in the piston crown that are necessary for figuring the temperature stresses may be determined from the condition that heat received by the piston from the gases is equal to the heat flow through the crown from the center to the cylinder walls. For simplicity the crown may be assumed flat and of constant thickness:

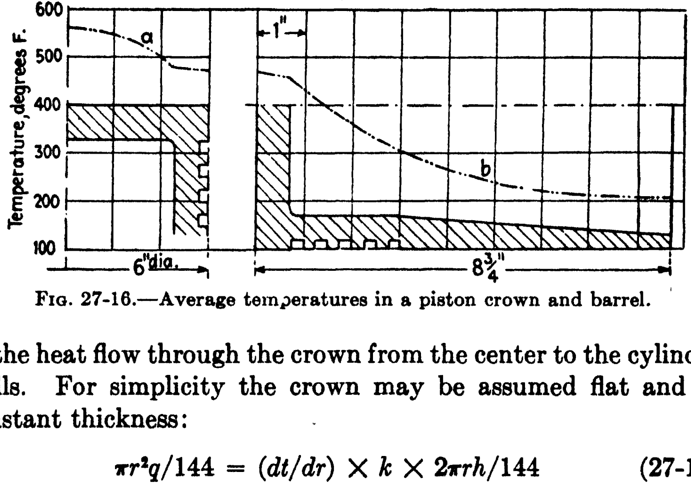


FIG. 27-16.—Average temperatures in a piston crown and barrel.

$$\pi r^2 q/144 = (dt/dr) \times k \times 2\pi r h/144 \qquad (27\text{-}15)$$

where r is the distance from the center to the cirumference of a circle concentric with the engine bore, in.,
q is the heat flow from the gases, Btu/sq ft-hr,
t is the temperature at the circumference with the radius r,
dt/dr is the temperature gradient,
k is the heat conductivity, Btu-in./sq ft-°F-hr
h is the thickness of the crown, in.

Equation (27-15) may be reduced to

$$dt = (q/2kh) r\, dr \qquad (27\text{-}16)$$

Integrating equation (27-16) from the center to the edge, where the temperature limits are t_c at the center and t_e at the edge and the limits of r are 0 and $D/2$, or half of the engine bore, gives

$$t_c - t_e = D^2 q/16kh \qquad (27\text{-}17)$$

Under the conditions set at the beginning, the temperature stresses are proportional to $(t_c - t_e)$. Temperature measurements made on pistons that were withstanding cracking indicate that a maximum permissible value of $(t_c - t_e)$ for cast iron is about 400 F and about half of this value for aluminum.

Thus, without determining the temperature stresses, the minimum thickness of the crown necessary to carry away the heat may be computed from the expression

$$h = D^2q/16k(t_c - t_e) \qquad (27\text{-}18)$$

The value of the heat flow q depends upon the piston material, mean effective pressure, and stroke-bore ratio. For cast-iron pistons in four-stroke engines it varies from 10,000 to 40,000 and in two-stroke engines from 30,000 to 80,000 Btu/sq ft-hr. For aluminum pistons q may be about two times higher.

The limits of D for uncooled pistons can be determined from expression (27-18), assuming a maximum value of 4 in. for h. It shows also that the introduction of aluminum with its higher k raises these limits.

If the crown is not flat but concave or convex, the conditions become more involved. However, it is more or less obvious that a concave or convex crown being less rigid can stand higher temperatures by changing its shape and thus, to a certain extent, equalizing the stresses. Figure 27-17 illustrates this point and at the same time explains why many designers advise against connecting the crown with the cylindrical part by ribs, as ribs make the construction rigid. There is, however, also a strong argument in favor of ribs as the ribs carry a considerable part of the heat from the center direct to the piston rings and to the liner, thus decreasing the temperature difference.

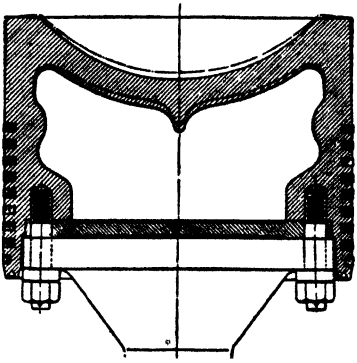


FIG. 27-17.—Heat distortion of a piston crown.

Oil-cooled Pistons.—Pistons of vertical engines are often cooled by oil instead of by water, because oil leakage into the crankcase is harmless, whereas water leakage forms an emulsion with the oil.

may cause corrosion, and in general is very undesirable. In large engines, oil is circulated in a similar way to water in water-cooled pistons. In medium-size and small engines the pistons are cooled either by admitting the oil through the hollow connecting rod to a closed space under the piston crown, Fig. 27-8, or by directing a jet of oil against the underside of the crown. In either case, the oil, after cooling the crown, flows down into the crankcase. The amount of heat carried away by the oil is about 5 per cent of the total heat dissipated by cooling.

Heat Expansion.—The temperature rise of some 300 to 400° in the crown of an uncooled trunk piston results in a considerable expansion of the piston. This expansion is greatest at the crown and decreases toward the end of the skirt with decrease of temperature, Fig. 27-16. Therefore the piston when cold must have a considerably smaller diameter than the cylinder bore. On the other hand, a trunk piston acts as a crosshead and takes the normal component of the forces acting on the connecting rod; if the difference between piston and cylinder diameter is too large, knocking and slapping will result.

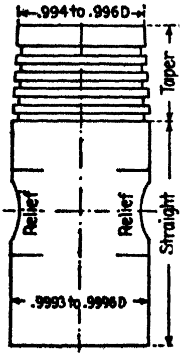


FIG. 27-18.—Allowances for piston expansion.

The amount by which a piston must be tapered depends naturally upon the highest temperature, coefficient of expansion of the material, and cylinder bore. Most designers make the barrel of the piston cylindrical from the end to the first piston ring, and from there on tapered. Figure 27-18 shows average practice for cast-iron pistons of oil engines. The crown of natural-gas engines may have a higher temperature, and the taper should be increased by about 50 per cent.

For aluminum pistons, the clearances between piston and cylinder should be twice as large. In airplane engines, which operate with much higher mean effective pressures and temperatures, still greater clearances should be provided.

Trunk pistons are stiffened by the wristpin bosses and have a tendency to expand more in the diameter coinciding with the wristpin, and, therefore, in order to prevent excessive friction and seizing, the sides of the pistons near the wristpin are relieved as shown in Fig. 27-18.

Cooled Pistons.—In water- or oil-cooled pistons the temperature of the crown is fairly uniform over the greater part of the area, with a gradual decrease toward the edge and down the sides; but even under these conditions the heat stresses in the crown are high, and in large trunk pistons the crown is made of a steel forging. The pistons of large double-acting two-stroke engines are very long, and special precautions must be taken to allow longitudinal expansion. Figure 27-13 shows an example of a large water-cooled piston for a two-stroke oil engine. Each half of the piston is made up from two main pieces fastened to the piston-rod collars by stud bolts. The gas pressures are transmitted directly to the piston-rod collars, a split central part *a* protecting the water connections from hot gases and lubricating oil.

27-4. Connecting Rods. *Materials.*—Connecting rods usually are forged of open-hearth steel, sometimes of nickel steel or vanadium steel. Those of high-speed engines are often made of duralumin and aluminum alloys. Connecting rods of low-speed horizontal gas engines sometimes are made of steel castings.

Shape of Rod.—In low-speed engines, the section of the rod is circular, with flattened sides; in high-speed engines either an H-section or tubular section is used because of their lightness. The rod usually tapers slightly from the big to the small end.

Strength.—The forces acting on the connecting rod are:

1. The joint effect of (*a*) the pressure on the piston combined with the inertia of the reciprocating parts; and (*b*) the friction of the piston rings, of the piston, piston rod, and crosshead.
2. The longitudinal component of the inertia of the rod.
3. The transverse component of the inertia of the rod.
4. The friction of the two end bearings.

The forces under 1(*a*) and 1(*b*) must be divided by the cosine of the angle of obliquity of the rod. The forces 1(*a*) are the net efforts, as discussed in Chap. 24. The friction of the piston rings may be determined from the expression

$$F = b\pi D i p_r f \tag{27-19}$$

where b is the width of the rings, in.,

D is the cylinder bore, in.,

i is the number of rings,

p_r is the actual pressure of the rings, 3.5 to 6 psi,

f is the coefficient of friction, about 0.1.

The friction of the piston and crosshead is produced by the normal component F_n of the piston pressure. The average value of F_n is small, not over 8 per cent of the piston pressure. With a friction coefficient of about 0.06, the friction force is less than 0.5 per cent of the piston pressure and therefore can be neglected.

The influence of the forces in item 1 is taken care of by determining the area A of the smallest section of the rod for compression, using an allowable stress of $S = 8000$ psi for mild steel up to 12,000 psi for nickel steel and vanadium steel. If the rod is made rectangular in section, the width b in the plane of rotation should be 1.5 to 1.8 times the thickness a. If the total pressure is F lb, then, designating $b/a = c$,

$$F/S = ab = ca^2 \tag{27-20}$$

and

$$a = \sqrt{F/Sc} \tag{27-21}$$

If the rod is round, an equation similar to (27-20) gives

$$d = \sqrt{F/0.7854S} \tag{27-22}$$

In a rectangular or H-section the thickness a is kept constant, increasing b toward the big end 20 to 25 per cent.

In checking the rod as a column in the plane of rotation, formula (27-13) may be used with $n = 1$, considering both ends hinged. The radius of gyration k in this formula refers to the section halfway between the two ends.

Item 2 is estimated with a sufficient accuracy by considering that the part of the rod from the small end to the center of gravity of the rod is reciprocating with the piston and the rest is rotating with the crank.

The transverse component of the sum of the centrifugal forces exerted by the different sections of the rod gives a bending moment which reaches a maximum when the connecting rod is tangent to the crank circle. Dividing this moment by the section modulus of the rod at the dangerous section, about 0.42 l from the big end, gives the bending stress s with a sufficient accuracy as

$$s = 0.00000182n^2rAwl^2/Z \tag{27-23}$$

where n is the engine speed, rpm,
r is the crank radius, in.,
A is the rod section, sq in.,
w is the specific weight of the metal, lb/cu in.,
l is the length of the rod, in.,
Z is the section modulus, in.^{3}

Length of Connecting Rod.—In modern engines the length varies from $l = 4r$, for high-speed engines, up to $l = 4.75r$, for low-speed engines. A few low-speed engines and most horizontal engines use $l = 5r$.

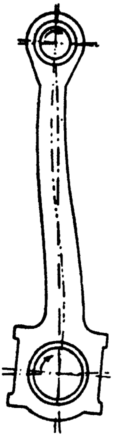


FIG. 27-19. Deflection by friction.

Polar load diagrams, Sec. 26-5, and determination of the minimum lubricating oil-film thickness h_m, Secs. 26-4 and 26-6, must be used in checking the length and diameter of the crankpin bearing.

Friction.—The friction in the two end bearings produces bending moments which tend to bend the connecting rod into an S-shape, shown exaggerated in Fig. 27-19. The maximum moments of resistance are at the sections where the rod shank joins the big and small ends.

For the big end the friction at the crankpin creates a reaction R at the piston pin normal to the connecting-rod center line

$$R = Ffr_c/l \tag{27-24}$$

where F is the maximum force acting on the connecting rod, lb,
f is the coefficient of friction,
r is the crankpin radius, in.,
l is the total length of the connecting rod, in.

The bending moment created by this reaction at the dangerous section, a in. from the crankpin center, Fig. 27-20, is, with expression (27-24),

$$M = R(l - a) = Ffr_c(l - a)/l \tag{27-25}$$

and the stress created by it, if the section modulus at this section is Z'

$$s' = Ffr_c(i - a)/lZ' \tag{27-26}$$

Similarly, the stress created by the friction of the wristpin is a maximum at a section b in. from the wristpin center, Fig. 27-21, and is

$$s'' = Ffr_w(l - b)/lZ'' \qquad (27\text{-}27)$$

where Z'' is the section modulus in bending of this section.

The stresses by the formulas (27-26) and (27-27) must be added to the direct compressive stresses. Normally the coefficient of friction is about $f = 0.02$ to 0.03; however, if lubrication begins to fail, f may go up to 0.05 and even higher. Therefore it is advisable to check at least for $f = 0.05$.

The allowable maximum crankpin-bearing pressure depends upon the bearing material, and method of lubrication. With babbitt and centrifugal-force lubrication from a ring fastened to the crankshaft the pressure should not exceed 1000 psi; with pressure lubrication it can be from 1200 to 1500 psi. With copper-lead bearings and pressure lubrication it can go up to 1800 psi and higher if the journal surface has a Brinell hardness of 300 or more. The data in Table 26-2 may serve as a guide. The use of higher pressures than given above will result in a faster bearing wear. A big relief r of the shells on the side, Fig. 27-25, prevents the closing of the bearing shells on the crankpin when the bearing begins to warm up through friction, ensures better lubrication, and thus considerably reduces the wear of the bearing.

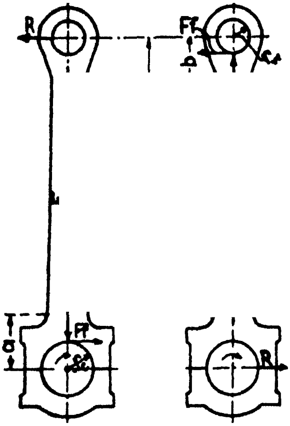


FIG. 27-20. FIG. 27-21.
Action of friction forces.

Connecting-rod Bearing Cap.—The cap must be not only strong but also as rigid as possible. It is difficult to give definite data about the permissible deflection, but the latter will be low if the maximum stress is kept low. It will be found that I-beam cross sections give the highest rigidity with the least weight.

The cap is designed assuming that it is a freely supported curved beam. The maximum force, taking into account both gas and inertia forces and using a polar diagram, is considered as applied to the center of the cap as a concentrated load; the supports are at the crank-bearing-bolt center lines. The bending moment on this beam is found for any section from the usual formula of a straight beam

$$M = 0.5Fx \tag{27-28}$$

where x is the distance from the center of gravity of the section

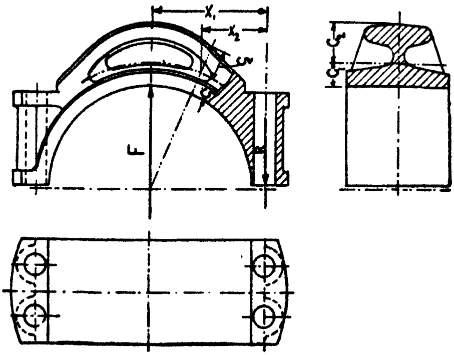


FIG. 27-22.—Connecting-rod cap.

under consideration to the nearest bolt, Fig. 27-22. The stress caused by the moment M is determined by using the straight-beam stress formula and applying a correction factor k. Thus for compression in the inner fibers

$$s_1 = k_1 M c_1 / I \tag{27-29}$$

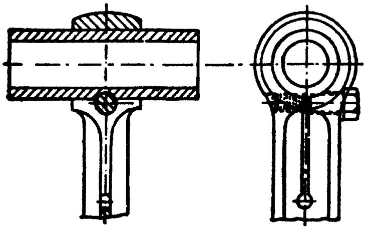


FIG. 27-23.—Small end of an automotive connecting rod.

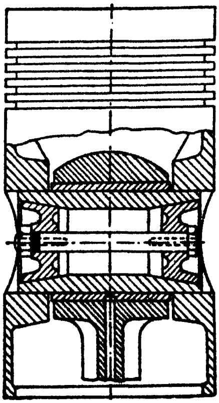


FIG. 27-24.—Wrist-pin with expander-type lock.

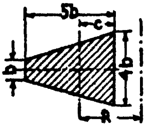
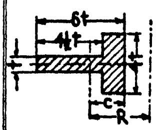
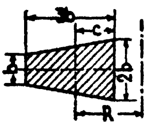
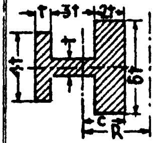
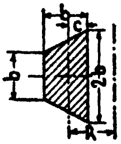
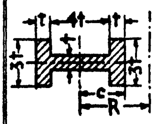
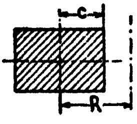
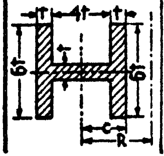
and the stress in tension in the outer fibers is

$$s_2 = k_2 M c_2 / I \tag{27-30}$$

The coefficients k_1 and k_2 may be found in Table 27-2; I is the moment of inertia of the section, and c_1 and c_2 are the distances from the inner and outer fibers, respectively, from the neutral axis.

Constructions.—In small high-speed engines, the small end clamps the piston pin, Fig. 27-23, and the big end has shims to take up the wear of the bearing. In some high-speed engines the piston pins are made full-floating-free to turn both in the connecting-rod bushing and in the piston bosses, Fig. 27-4. In larger engines the piston pin is fastened in the piston, and the small end is equipped with a bronze bushing. Figure 27-24 shows an expander-type locking device. In still larger engines the big end has a separate bearing and, in the case of high-compression oil engines, shims s, Fig. 27-25, to adjust compression. Figure 27-8 shows a connecting rod bolted to the piston pin.

TABLE 27-2.—CONSTANTS FOR CURVED BEAMS

Section	$\frac{R}{c}$	Constants		Section	$\frac{R}{c}$	Constants	
		Inside fibers k_1	Outside fibers k_2			Inside fibers k_1	Outside fibers k_2
	2.0	1.61	0.68		2.0	1.73	0.72
	3.0	1.34	0.76		3.0	1.41	0.79
	4.0	1.24	0.82		4.0	1.29	0.83
	6.0	1.15	0.87		6.0	1.18	0.88
	8.0	1.12	0.91		8.0	1.13	0.91
	2.0	1.61	0.73		2.0	1.69	0.80
	3.0	1.37	0.81		3.0	1.38	0.86
	4.0	1.26	0.86		4.0	1.26	0.89
	6.0	1.17	0.91		6.0	1.15	0.92
	8.0	1.13	0.94		8.0	1.10	0.94
	2.0	1.58	0.71		2.0	1.41	0.79
	3.0	1.33	0.80		3.0	1.23	0.86
	4.0	1.23	0.84		4.0	1.16	0.89
	6.0	1.13	0.88		6.0	1.10	0.92
	8.0	1.10	0.91		8.0	1.07	0.94
	2.0	1.52	0.73		2.0	1.36	0.83
	3.0	1.30	0.81		3.0	1.19	0.88
	4.0	1.20	0.85		4.0	1.13	0.91
	6.0	1.12	0.90		6.0	1.08	0.94
	8.0	1.09	0.92		8.0	1.06	0.95

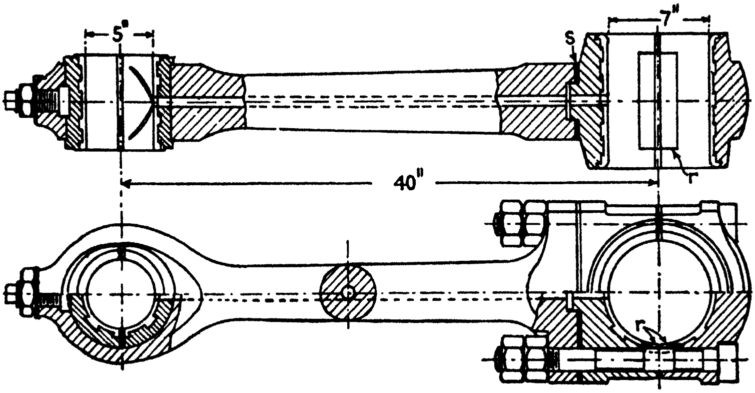


FIG. 27-25.—Connecting rod with adjustable bearings.

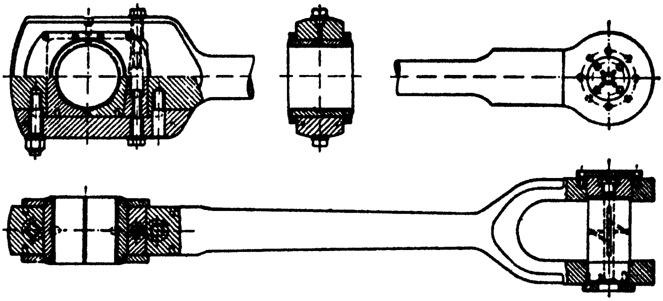


FIG. 27-26.—Connecting rod for a large horizontal engine.

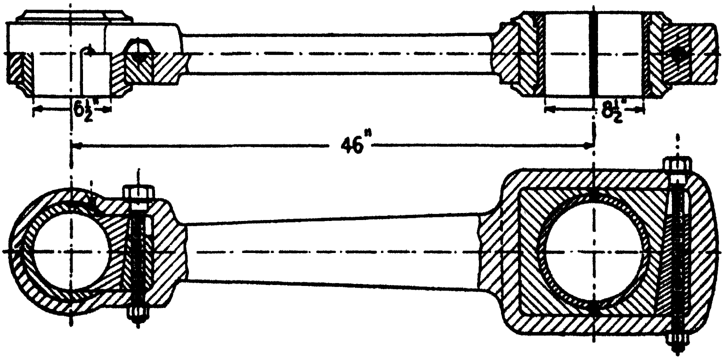


FIG. 27-27.—Connecting rod for a horizontal gas engine.

A rod with a forked small end for use with a crosshead and wedge take-up of the big-end bearing is shown in Fig. 27-26 and a connecting rod with wedge take-up at both ends as used in large double-acting gas engines with a side crank is shown in Fig. 27-27.

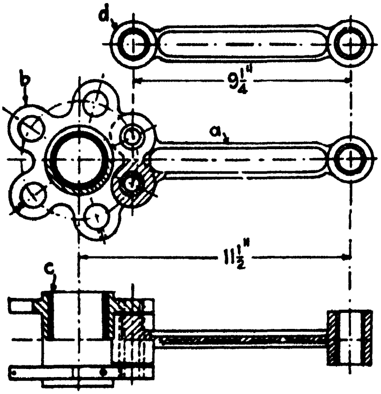


FIG. 27-28.—Parts of a five-cylinder radial engine.

Finally, Fig. 27-28 shows parts of a radial airplane engine: a master connecting rod *a*, a cluster *b* which is its big end with a babbitt-lined bronze bushing *c* as bearing, and one of the auxiliary, or *articulated*, rods *d*.

Connecting-rod Bolts.—The crankpin bearing is fastened to the rod by two or four bolts. In four-stroke engines these bolts are designed to take the maximum inertia load with a small stress of 5000 to 6000 psi based on the inertia and centrifugal loads divided by the area of the bolts at the bottom of the threads. It is difficult to avoid failure of these bolts in trunk-piston engines when due to piston seizure.

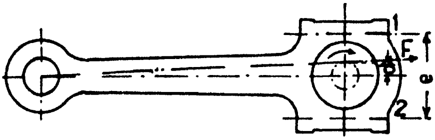


FIG. 27-29.—Friction action on big-end bolts.

The bolts are not stressed equally: Owing to the journal friction, the pull in bolt 1, Fig. 27-29, is greater and equal to

$$F_1 = F(e/2 + d)/e \qquad (27\text{-}31)$$

where *d* is the arm of the force *F* in the moment expression (27-25)

$$d = fr_e(l - a)/l \qquad (27\text{-}32)$$

With crosshead engines the small-end bolts have to carry the inertia load due to piston, piston rod, and crosshead and also friction forces acting on the piston and crosshead. As these forces are rather indeterminate, it is advisable to allow about 30 per cent less stress on these bolts than on the big-end bolts.

Connecting-rod bolts of two-stroke engines are made of the same size as those for four-stroke engines, as there is always a possibility of loss of compression resulting in heavy inertia loads.

Crankpin-bearing bolts may be subject to considerable repeated stresses in the case of torsional vibration of the crankshaft, particularly if the crankpin bearing is separate from the connecting rod as in Fig. 27-25. Spigoting the bearing into the foot of the rod reduces the stress but may not be sufficient to avoid failure. For this reason some engine manufacturers advise the change of the bolts at regular intervals, once every 12 to 18 months, as the cost of bolts is slight compared with the consequences of bolt failure while the engine is running.

A better practice is to inspect the bolts periodically after a certain number of hours of engine operation, 3000 to 5000, by taking them out and painting them with chalk dissolved in alcohol. If a crack has been started, the oil from the crack will turn the chalk over the crack yellow. Such a crack is the beginning of a progressive fracture, and the bolt must be exchanged.

27-5. Problems.—1. Find the thickness of the head of a cast-iron oil-engine trunk piston 12 in. diameter, the maximum pressure is, 550 psi.

2. Find the thickness of the head of a cast-iron gas-engine trunk piston, 10 in. diameter; the maximum pressure is 375 psi.

3. Find the thickness of the head of a heavy-duty oil-engine piston if the diameter is 8 in., the maximum pressure is 1250 psi, and if made of: (*a*) cast iron, (*b*) cast aluminum alloy, and (*c*) an aluminum forging.

4. Find the theoretical thickness of the head of an aluminum automobile-engine piston 2¾ in. diameter, the maximum pressure is 400 psi. Compare this with the thickness computed from an empirical formula.

5. Determine the stress in the crown of a cast-iron gas-engine trunk piston 14¼ in. diameter, 1½ in. thick, 320 psi pressure.

6. Determine the stress in the crown of an aluminum oil-engine trunk piston 6 in. in diameter, 1⅛ in. thick; the maximum pressure is 700 psi.

7. Determine the stress in the crown of a cast-iron oil-engine flat-top trunk piston. The diameter is 8¾ in., the thickness is 1⅜ in.; the maximum pressure is 1000 psi.

8. Determine the stress in the head of a cast-iron automobile-engine piston. The diameter is 3⅜ in.; the thickness is 0.19 in. at the center increasing to 0.27 in. near the juncture with the barrel; the maximum pressure is 360 psi.

9. Determine the stress in the crown of an aluminum flat-top truck-engine piston. The diameter is 4⅝ in., the thickness is 1 in., the maximum pressure is 950 psi.

10. Determine the thicknesses of the barrel at the head, just below the piston rings, and at the bottom, also the thickness and width of the piston rings for the trunk piston in problem 1.

11. Work problem 10 for the piston of problem 2.

12. Work problem 10 for the piston of problem 3.

13. Work problem 10 for the piston of problem 5.

14. Work problem 10 for the piston of problem 6.

15. Work problem 10 for the piston of problem 7.

16. Design the wristpin for the piston in problem 1, for both bearing pressure and bending stresses.

17. Design the wristpin for the piston in problem 1, using construction and proportions shown in Fig. 27-7.

18. Design the wristpin for the piston of problem 7 using the construction and approximate proportions shown in Fig. 27-8.

19. Design the piston rod for a double-acting gas engine, bore 22 in., stroke 30 in., maximum pressure 360 psi. Compare the weight of a solid rod with that of a hollow one with the inside diameter equal to one-third of the outside diameter. Assume the length of the rod from the piston to the crosshead 1.7 times the stroke.

20. Design the hollow piston rod for a double-acting four-stroke oil engine; the bore is 9 in., the stroke is 12 in., the maximum pressure is 900 psi; the engine speed is 720 rpm. Assume the weight of the reciprocating masses as 15 psi; make the inside diameter of the rod $0.35d$, where d is its outside diameter; the length of the rod between the supports is 22 in.

21. Determine the axial expansion of the piston rod of problem 20 if the water flowing through the rod has a mean temperature of 165 F and the outside surface of the rod has a temperature of 400 F.

22. Find the temperature stress in the piston rod of problems 20 and 21 taking into account both the axial and radial expansion.

23. Determine the axial expansion of a hollow piston rod 14 ft 6 in. long, 8¼ in. outside and 3 in. inside diameter; the mean cooling water temperature is 172 F, the average heat flow is 23,000 Btu/sq ft-hr.

24. Find the temperature stress in the rod of problem 23 taking into consideration both the axial and radial expansion.

25. Determine the size of a crosshead shoe for the engine in problem 19 using length 1.5 times the width.

26. If a temperature difference of 200 F creates in a certain flat cast-iron piston crown a stress of 5500 psi, determine the temperature stress in a 6-in. piston with a ⅞ in. thick crown and a maximum heat flow of 20,000 Btu/sq ft-hr.

27. Determine the size of a crosshead shoe for the engine in problem 20 using a length 1.4 times the width.

28. Find how much the temperature stress in problem 26 will be changed if the piston is made of an aluminum alloy; (*a*) keeping the same dimensions and (*b*) using forged aluminum that permits a reduction of the thickness to ⅝ in.

29. Determine the amount of clearance between the crown of an 8-in. aluminum uncooled piston and the cylinder liner, at full load. The temperatures of the piston are the same as given in Fig. 27-16 and of the cylinder liner in Fig. 22-12. At room temperature the crown is 0.75 per cent smaller than the cylinder bore.

30. Determine: (*a*) the thickness of the piston crown in problem 29 which will give the temperature differences indicated in Fig. 27-16; (*b*) the heat flow; and (*c*) the maximum temperature stress for that piston.

31. Design a connecting rod for the engine in problem 1, assuming conventional ratios of stroke to bore and rod length to crank radius, making the rod section round and taking into consideration all acting forces.

32. Determine the stress due to whipping in the connecting rod from problem 31, if the engine speed is 310 rpm.

33. Design a steel connecting rod with an I-section for the engine in problem 4, indicating the necessary assumptions.

34. Design an aluminum-alloy connecting rod for the engine in problem 4, making the same assumptions.

35. Compare the factors of safety of the connecting rods as found in problems 33 and 34 when considering the forces due to whipping at 2000 rpm.

36. Determine the size of bolts necessary for fastening the wristpin in problem 17 to the connecting rod, using four bolts and a coefficient of friction $f = 0.05$.

37. Determine the main dimensions of a forged-steel connecting rod with an I-section for the engine of problem 7. Select a suitable connecting-rod length to crank radius ratio; the engine stroke is 12 in., the speed 720 rpm.

38. Determine the suitable cross section of the large-end bearing cap for the connecting rod of problem 37.

39. Determine I-section dimensions for the connecting rod for the engine of problems 7 and 37, taking into account the stresses created due to whipping of the rod.

40. Determine the size of the bolts for the connecting-rod cap of problems 37 and 39.

41. Determine the bending stresses due to journal friction in the connecting rod of problem 31, assuming the crankpin diameter equal to 1.5 times that of the wristpin.

42. Determine the size of the crankpin bearing bolts for the rod in problems 31, 32, and 41. Assume the weight of the reciprocating parts from Table 24-3.

43. Determine the bending stresses due to journal friction in the connecting rod of problems 37 and 38; make the crankpin diameter 1.75 times that of the piston pin.

CHAPTER 28

CRANKSHAFTS

28-1. General. *Types.*—All crankshafts may be divided into two types—shafts with side cranks and shafts with center cranks. The latter constitute the majority by far and may be further subdivided into single-crank and multicrank shafts. So far as the design is concerned a very important factor is the number of bearings. Center crankshafts of larger engines generally have a bearing between each pair of cranks and a bearing on each shaft end. In automobile engines, in order to make the engine shorter and lighter, the number of bearings may be decreased. Thus a four-throw crankshaft may have either five or three or even two bearings; a six-throw crankshaft may have seven, four, or three; and so on.

In V-type engines each crankpin serves two cylinders, and in radial engines one crankpin serves all cylinders—three, five, seven, or nine.

Firing Order.—To secure uniform rotation in multicylinder engines, the power impulses should be equally spaced. This consideration in connection with balance requirements determines the conventional arrangements of crankshaft throws and firing orders as shown in Fig. 28-1. The cylinders are numbered starting either from the flywheel end or, as in automobile engines and some large oil engines, from the forward end toward the flywheel. Looking at the flywheel, most engines, including automobile engines, rotate counterclockwise.

Materials.—Crankshafts of stationary engines are usually made of open-hearth steel with a tensile strength of 68,000 psi, an elastic limit of about 38,000 psi, and an elongation of 25 per cent in 2 in. The crankshafts of marine oil engines are made of steel with a smaller carbon content, 62,000 to 66,000 psi tensile strength and an elongation of 30 to 28 per cent, respectively. Automobile- and airplane-engine crankshafts are usually made of chrome-nickel steels which, after heat-treating, have a tensile strength of from 130,000 to 150,000 psi, elastic limits of about 20 to 30 per cent below the ultimate strength, and an elongation of about 16 to 20 per cent.

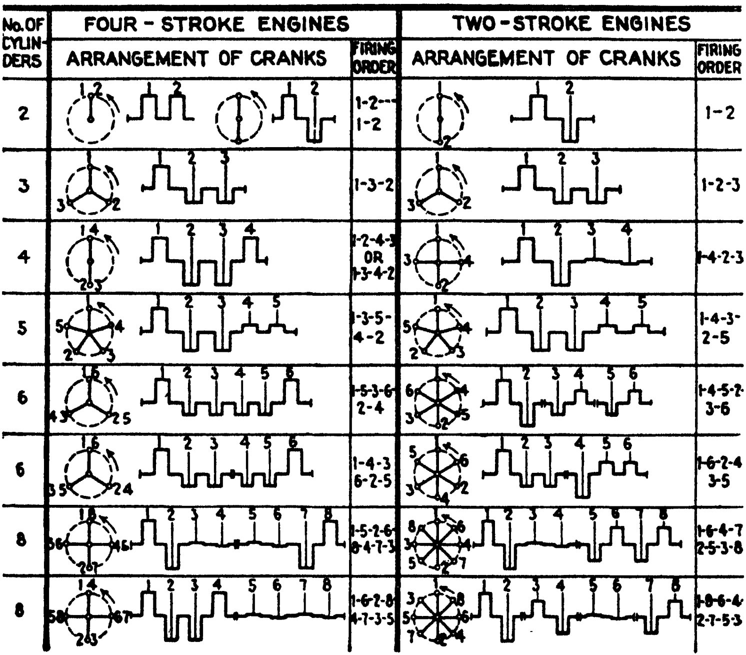


FIG. 28-1.—Arrangement of cranks and firing-order diagram.

Table 28-1 gives the specifications of some typical materials.

TABLE 28-1.—SPECIFICATIONS OF CRANKSHAFT MATERIALS

Material	Ultimate tensile strength S_u	Elastic limits		Endurance limits		Moduli of elasticity	
		Tension S_e	Shear S_{es}	Bending S_{en}	Torsion S_{ens}	Direct E	Transverse G
	psi	psi	psi	psi	psi	psi	psi
Open-hearth steel	68,000	38,000	24,000	27,000	20,000	30,000,000	11,500,000
Alloy steel	85,000	60,000	36,000	40,000	24,000	30,500,000	12,500,000
Cast steel, mild	80,000	32,000	20,000	32,000	18,000	29,000,000	11,600,000
Cast steel (1.4–2.0 Cu)	86,000	35,000	22,000	34,000	20,000	29,000,000	11,600,000
Cast iron, alloy	52,000	30,000	30,000	18,000	18,000	20,000,000	8,000,000

Crankshafts of many automobile engines and some heavy-duty oil engines are made of special cast-iron alloys, Table 28-2.

TABLE 28-2.—SPECIFICATIONS OF CAST CRANKSHAFTS

Constituents	Small engines[a]	Large engines[a]	Large compr.-ign. oil engines[a]	Heavy-duty oil engines	Automobile engines
Total carbon, %	2.75	3.00	2.50	2.30–2.80	0.40–0.60
Combined carbon, %	0.75–1.00	0.80–1.00	0.45–0.70	0.40–0.75	
Silicon, %	1.90	2.30	2.20	1.75–2.50	0.90–1.10
Sulphur, max, %	0.08	0.08	0.05	0.10	0.08
Phosphorus, max, %	0.20	0.20	0.05	0.25	0.20
Manganese, %	0.75	0.55	. . . .	0.75–1.25	0.60–0.80
Nickel, %	1.50	1.00	1.00	0.75–2.00	
Molybdenum, %	0.75	0.50	1.00	0.75–1.35	Cr. 0.50
Copper, %	. . . .	. . . .	. . . .	. . . .	1.40–2.00
Ultimate tensile strength, min, psi	55,000	55,000	50,000	52,000	86,000
Brinell hardness number	220–270	220–270	285–300	269–325	255–302

[a] *Product Eng.*, June, 1935.

The use of cast-iron alloys for crankshafts is steadily increasing, particularly for large and high-speed engines. Casting of the crankshaft permits a theoretically desirable but complicated shape with a minimum of machining and at least cost. In addition, cast iron has the advantage of a considerably smaller sensitivity to notch effect than forged steel.

Manufacturing.—The crankshaft is one of the most important parts of an engine, and the greatest care should be taken in its manufacture. Smaller crankshafts are drop forged; larger shafts must be forged. Cast-iron crankshafts are cast in permanent molds which ensure a maximum accuracy and a minimum of machining. When machining crankshafts, special precautions must be taken to avoid any springing; the shaft should be well supported between the centers. After turning, the journals and crankpins are ground to exact size and perfect smoothness. Finally the crankshafts are balanced—large shafts of low-speed engines are balanced statically; crankshafts of high-speed engines are balanced dynamically on special balancing machines.

Bearing pressures are the first thing to be considered in designing a crankshaft. The permissible pressures depend upon the maximum cylinder pressure, change of direction of the bearing pressure, journal velocity, and method of lubrication. Maximum bearing pressures, computed for a great number of different engines operating satisfactorily, are given in Table 26-2 together with the method of their lubrication and other lubricating factors. In all four-stroke and double-acting two-stroke engines the pressure is relieved from the bearings at least once during each cycle. This pressure release is very important in restoring the oil film, which may be forced out by an excessive pressure. In two-stroke single-acting engines the pressure always acts in one direction, and as a result the maximum permissible pressures cannot be as high as in single-acting four-stroke engines.

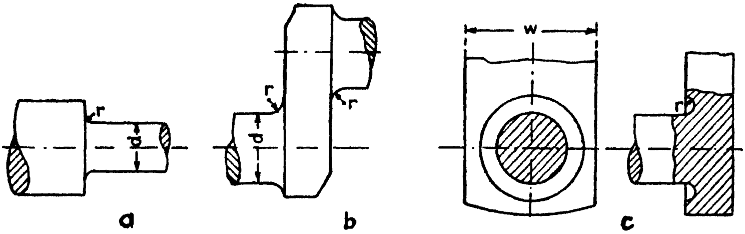


FIG. 28-2.—Use of fillets in junctures of different sections.

Stresses.—The stresses which arise in a crankshaft are from bending and twisting moments. As the failure of a crankshaft is apt to cause a serious engine wreck, and also because not all of the acting forces can be measured accurately, the allowable stress limits usually are taken rather low. For open-hearth steel, 6000 to 7000 psi is the limit for bending where the stress is reversed and 10,000 to 12,000 psi where it is acting in one direction only; 5000 to 6000 psi is the usual limit for the shear stress, and 12,000 psi is the limit for combined stresses. In cast-iron shafts, Table 28-1, the allowable stresses are about 25 per cent lower. As a result, cast-iron shafts in heavy-duty compresssion-ignition oil engines have shaft diameters up to 0.8 of the cylinder bore.

High stress concentration may occur as a result of abrupt changes of section, unsymmetrical drilled holes or sharp-ended keyways. Two different cross sections must be blended with a large fillet r, Fig. 28-2*a* or 28-2*b*. If possible, the radius r

should not be less than $0.2d$. If there is no room for the required fillet, it is advisable to undercut the cheek to obtain the fillet r, Fig. 28-2c, and to avoid a weakening of the cheek by increasing the width w. Figure 28-3 shows such undercut fillets in a built-up crankshaft for a large gas engine.

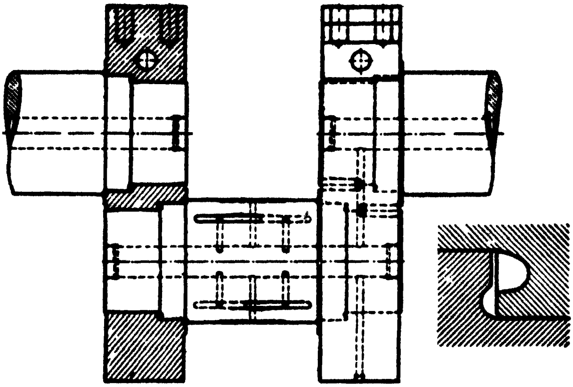


FIG. 28-3.—Built-up crankshaft. Scale 1:46.

Endurance Failure.—Most crankshaft failures are due to progressive fracture either from repeated bending or from reversed torsional stresses. Stresses repeated many times result in final failure of the stressed part, even though the stresses do not reach the elastic limit of the material. The maximum stress that can be applied indefinitely without causing failure is called the *endurance limit.* For open-hearth steel used for crankshafts the endurance limit for bending is about 0.43 of the ultimate strength, or 32,000 psi. For chrome-nickel and other alloy steels it is correspondingly higher. The endurance limit for shear is about half of the one for bending.

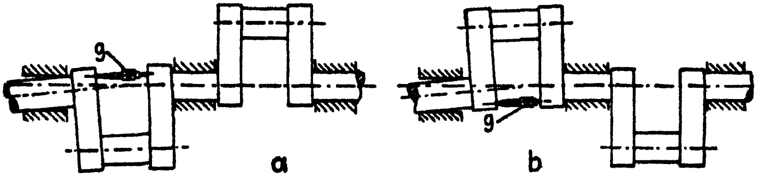


FIG. 28-4.—Checking alignment with a micrometer.

Repeated bending stresses exceeding the endurance limit may be produced in a crankshaft from one or more bearings being lower or higher than the rest. The easiest way to detect such a bearing is by using a micrometer gage *g* and checking the distance between the cheeks in several positions of the crank, as shown exaggerated in Fig. 28-4.

An overlap *a*, Fig. 28-5*a*, of the journals and crankpins increases the endurance strength materially, up to 50 per cent, as compared with shafts having a negative overlap *b*, Fig. 28-5*b*.[1] Nitriding of the journals increases the endurance strength up to 50 per cent, whereas chrome plating and surface hardening, particularly by high-frequency induction heating and quenching, reduce the endurance strength up to 40 per cent.[1] Cutting off the web corners, Fig. 28-5*a* and *b*, gives a more even force flow and reduces stress concentration at the junctures of the journals and webs, thus further increasing the endurance strength of the crankshaft as a whole.

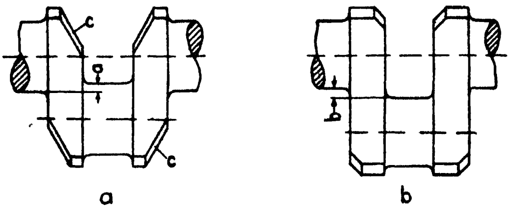


Fig. 28-5.—Overlap of journals and crankpin.

Repeated dangerous torsional stresses may occur in multithrow crankshafts from torsional vibration at critical speeds. In single-throw and two-throw crankshafts these stresses may occur from counterweights placed into the rim of the flywheel instead of being fastened to the crank cheeks. In a multithrow nonrigid crankshaft, with a heavy flywheel on one end and the power take-off on the other end, the torsional stresses may exceed the endurance limit at other than the critical speeds.

Torsional vibrations act as shock loads, and the resulting stresses are twice as high as when produced by static loads with the same angle of torsion. The allowable shear stresses are not over 0.36 S_{en}, better, $0.32S_{en}$, where S_{en} is the endurance limit in shear, about 20,000 psi for open-hearth steel and 18,000 psi for cast irons, Table 28-1.

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[1] *Engineering*, July 24, 1942, p. 78.

Figure 28-6 shows the consecutive steps of the development starting with a solid steel forging and ending with the ideal shape obtainable only with cast iron.[1] Figure 28-7 shows a cast-iron crankshaft for a small high-speed diesel engine. In spite of the compact design the shapes advantageous for cast-iron crankshafts are here successfully realized. Figure 28-8 shows a crankshaft of an automobile engine. The relatively great length of the crankshaft permits a very smooth force flow.

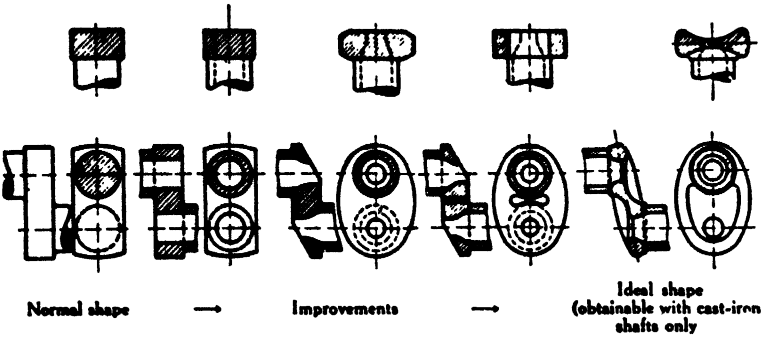

FIG. 28-6.—Development in the shaping of crankshafts.

Heat Expansion.—The expansion of crankshafts both from the heat radiated by the pistons and from heat of friction in the bearings must be met by allowing sufficient clearances between the crank cheeks and bearing shells. This elongation may be considerable.

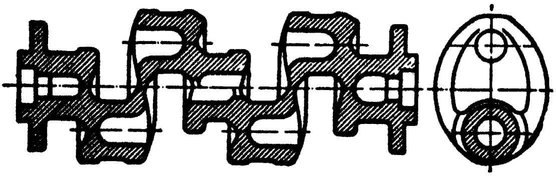

FIG. 28-7.—Cast iron crankshaft of a high-speed diesel engine of small output. (*Courtesy of Sulzer Bros., Ltd.*)

Example 28-1.—Find the expansion of a crankshaft 10 ft long with a temperature increase of 110 F over the room temperature.

Applying expression (21-17) with $a = 6.5 \times 10$, Table 21-1, the total expansion is

$$l_2 - l_1 = 10 \times 12 \times 6.5 \times 10 \times 110 = 0.086 \text{ in.}$$

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[1] MARTINAGLIA, L., Structural Durability of Crankshafts, *Sulzer Tech Rev.*, No. 2, 1943, pp. 19–28.

In order to prevent excessive lengthwise play, the crankshaft should be fixed in one of the middle bearings, making the distance between the cheeks of this bearing to fit the bearing shells and giving a necessary clearance at all the other bearings.

Determination of Dimensions.—First, the magnitudes of the various loads on the crankshaft must be determined. Next must be determined the distances between supports and the location of the loads in reference to the supports. For the sake of simplicity and also safety, the shaft is considered supported at the centers of the bearings. The distances between supports depend on the lengths of the bearings, and the latter depend on the diameters of the shaft on account of the permissible bearing pressures. Therefore, certain maximum bearing pressures and the length-to-diameter ratios, based on available data, Table 26-2, are assumed and these two dimensions determined according to the acting loads. These dimensions must be checked for the allowable bearing characteristic number Zn/p and film thickness h_m, as explained in Chap. 26. After that the thickness of the cheeks is assumed, about 0.5 d to 0.6 d, if d is the shaft diameter. With these values the distances between the supports are determined, and assuming allowable bending and shearing stresses the necessary dimensions of the crankshaft are determined.

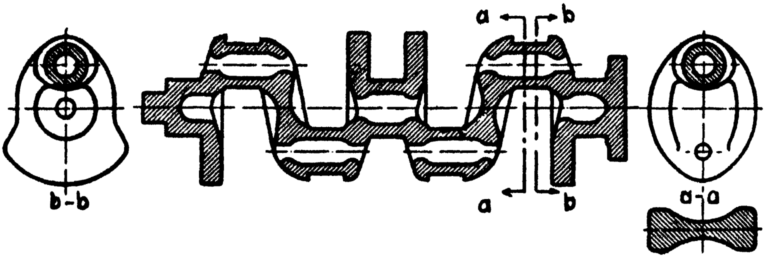


FIG. 28-8.—Cast-iron crankshaft of an automobile engine.

All forces and reactions are assumed to act at the centers of the bearings.

When computing the bending moments at various points of a crankshaft, two methods may be used. One method is to find the net reactions at the supports due to all the loads and then to determine the moments in various points. Another method is to compute the reactions and bending moments due to each load separately and then to combine all the moments graphically or analytically.

28-2. Center Crankshaft.—The crankshaft must be designed or checked for at least two crank positions—one when the bending moment from the gas pressure is a maximum, the other when the twisting moment is a maximum. In both positions the additional moments due to the flywheel weight and to the belt tension also must be considered.

To make the computations simpler without losing much of their accuracy it is customary to assume that the influence of bending forces does not extend beyond the two bearings between which a force is applied.

Crank on Dead Center.—The three-bearing crankshaft of a horizontal engine with its loads and distances of their application at this position is shown in Fig. 28-9.

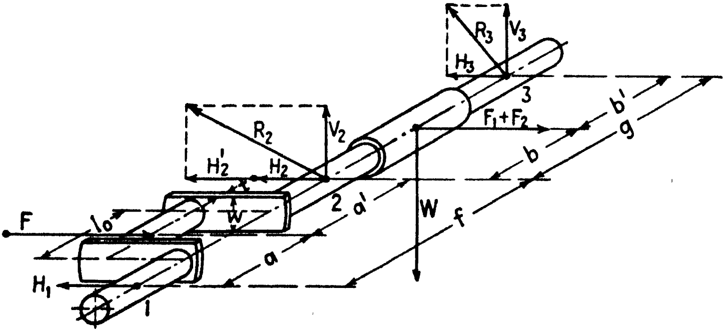


FIG. 28-9.—Forces acting on a crankshaft at dead center.

The gas pressure acting upon the piston gives force F resolved into two horizontal reactions at the bearings 1 and 2,

$$H_1 = Fa'/f \qquad \text{and} \qquad H_2 = Fa/f \tag{28-1}$$

The weight of the flywheel W gives two vertical reactions at the bearings 2 and 3,

$$V_2 = Wb'/g \qquad \text{and} \qquad V_3 = Wb/g \tag{28-2}$$

The belt pull $F_1 + F_2$ gives horizontal reactions,

$$H_2' = (F_1 + F_2)b'/g \qquad \text{and} \qquad H_3 = (F_1 + F_2)b/g \tag{28-3}$$

As stated before, the reactions H_1 and H_2 are considered as bending forces for the shaft piece between the bearings 1 and 2 and the reactions H_2' and H_3 between the bearings 2 and 3. However, for the pressure on the bearing 2 all three reactions must be combined, giving the reaction R_2. The axial dimensions must be determined first or assumed, as explained above, and then the diameters and crank dimensions may be determined.

Crankpin.—The bending moment at the center of the crankpin is

$$M = H_1a \tag{28-4}$$

and the diameter of the crankpin must be

$$d \geqq \sqrt[3]{32M/\pi S} \tag{28-5}$$

where the allowable stress S may be 10,000 to 12,000 psi.

Left-hand Crank.—The thickness t of the crank arm is made from $0.22D$ to $0.32D$, where D is the cylinder bore. The necessary width w of the crank arm may be determined from the bending moment on the arm

$$M = H_1(a - l_0/2 - t/2) = s_1Z \tag{28-6}$$

where the section modulus

$$Z = wt^2/6 \tag{28-7}$$

The stress due to direct compression is

$$s_2 = H_1/wt \tag{28-8}$$

and the total stress $s = s_1 + s_2$ should not exceed the allowable stress S' of 10,000 psi.

Right-hand Crank.—The bearings 1 and 2 are usually of the same length and symmetrical to the cylinder center line; this gives $H_1 = H_2$ and the same dimensions t and w for both cranks.

Shaft under Flywheel.—The bending moment due to the flywheel weight is

$$M_f = V_3b' = Wbb'/g \tag{28-9}$$

The moment due to the belt pull is

$$M_b = H_3b' = (F_1 + F_2)b'b/g \tag{28-10}$$

As one moment is in the vertical and the other in the horizontal plane, the combined moment is

$$M = \sqrt{M_f^2 + M_b^2} \tag{28-11}$$

The minimum diameter of the shaft must be

$$d = \sqrt[3]{32M/\pi S} \tag{28-12}$$

where the allowable stress S is 6000 to 7000 psi to take care of the reversal of stresses with each revolution and also to ensure the necessary rigidity.

If the crankshaft under the flywheel transmits a torque, the additional shear stress is taken into account as indicated in connection with equation (28-13).

Crank at Angle of Maximum Torque.—The forces acting on the shaft at a position where the torque or twisting moment is a maximum are shown in Fig. 28-10. The crank angle α, where the maximum torque occurs, may be found graphically by taking the pressures from the net-effort diagram for different crank angles and determining the tangential components F_t as explained in Sec. 24-4. That crank angle which corresponds to the largest value of F_t is the angle of the maximum torque. This angle usually lies between 25 and 35° from the corresponding dead center for engines with the combustion approaching constant volume and between 30 and 40° for engines with a constant-pressure combustion.

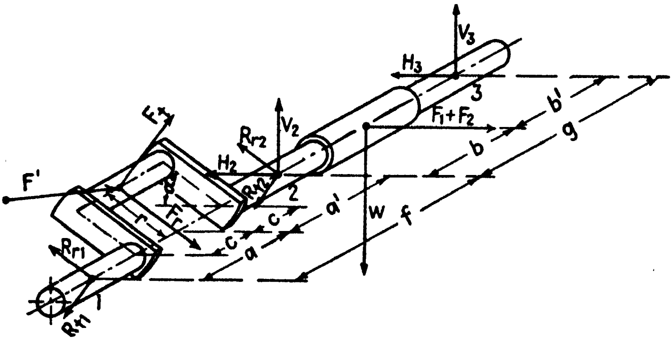


FIG. 28-10.—Forces acting on a crankshaft at angle of maximum torque.

The reactions created by W and $F_1 + F_2$ at the three bearings are the same as at the first position, and those due to the force F' exerted by the connecting rod on the crankpin can be resolved into reactions R_{r1} and R_{r2}, parallel to the radial component F_r and reactions R_{t1} and R_{t2}, parallel to the tangential component F_t, as shown in Fig. 28-10.

In this position the different crankshaft sections are subjected to bending and torsional moments and must be checked for the combined stresses. The two more generally used methods of finding the combined stresses will be discussed. The first method consists in finding separately the tensile stress s from bending and the shear stress s_s from torsion and computing the combined stress s' from the expression

$$s' = 0.35s + 0.65\sqrt{s^2 + s_s^2} \qquad (28\text{-}13)$$

Expression (28-13) is based on the theory that the safety of the shaft depends chiefly on the maximum deformation or the greatest tensile stress and therefore the combined stress s' should not exceed the allowable tensile stress of about 10,000 psi.

The second method is based on the theory that failure begins when the shear stress attains a certain value. For steel used in internal-combustion engine shafts this critical value is about 12,000 psi, and the allowable shear stress is therefore about 5000 or 6000 psi. With this method the bending moment M and the twisting moment T are combined into an equivalent twisting moment T_e, where

$$T_e = \sqrt{M^2 + T^2} \qquad (28\text{-}14)$$

The combined stress due to this moment is found as a shear stress s_s, and

$$s_s = T_e/Z_o \qquad (28\text{-}15)$$

where Z_o is the polar section modulus, and for a round shaft

$$Z_o = \pi d^3/16 \qquad (28\text{-}16)$$

The first method is used chiefly by European designers, whereas the second method is used by American and British designers.

Crankpin.—The bending moment M is found similarly to the moment at dead center, expression (28-4),

$$M = a\sqrt{R_{r1}^2 + R_{r2}^2} \qquad (28\text{-}17)$$

Using the diameter d found from expression (28-5), the combined stress may be found from expression (28-13) or (28-15). If the stress exceeds the allowable values as given for S or S_s, respectively, the preliminary value of d must be increased correspondingly.

Left-hand Crank.—This crank is subjected to bending due to R_{r1}, to direct compression due to R_{r1}, and to torsion due to R_{t1}. However, this crank does not transmit any power and has the same dimensions as the right-hand one which transmits the power to the flywheel and to the power take-off; therefore only the right-hand crank needs to be checked.

Right-hand Crank.—This crank is subjected to bending stresses in two planes normal to each other, to direct compression, and to torsion.

The bending moment due to R_{r2} is constant throughout the length of the crank and is

$$M_r = R_{r2}(a' - l_0/2 - t/2) \tag{28-18}$$

and the corresponding stress is

$$s_1 = M_r/Z \tag{28-19}$$

where the section modulus is

$$Z = t^2w/6 \tag{28-20}$$

The bending moment due to the tangential component R_{t2} is a maximum at the juncture of the crank and shaft and is

$$M_t = F_tr' \tag{28-21}$$

where the distance

$$r' = r - 0.5d_2 \tag{28-22}$$

d_2 being the shaft diameter at bearing 2. The resulting stress is

$$s_2 = M_t/Z' \tag{28-23}$$

where the section modulus

$$Z' = tw^2/6 \tag{28-24}$$

The stress in direct compression is

$$s_3 = R_{r2}/wt \tag{28-25}$$

The maximum stress will be in compression on the upper left corner of the cross section of the crank and equal to the sum of the above stresses

$$s_c = s_1 + s_2 + s_3 \tag{28-26}$$

The torsional moment is

$$T = R_{t2}(a' - c) \tag{28-27}$$

and the corresponding shear stress is

$$s_s = T/Z_o \tag{28-28}$$

where the polar section modulus is

$$Z_o = t^2w/4.5 \tag{28-29}$$

The total stress s_{max} is found by combining s_c and s_s by means of expression (28-13). If s_{max} exceeds 10,000 psi, the crank cheek is generally strengthened by increasing the width w, because this increase does not affect any other dimensions.

Juncture of Crank and Shaft.—The bending moment here is

$$M = H_2(a' - 0.5\,l_0 - t) \tag{28-30}$$

and the torsional moment is

$$T = F_tr \tag{28-31}$$

These two moments can be combined using expression (28-14), and the shear stress due to the equivalent torque should not exceed 5000 to 6000 psi.

Shaft under Flywheel.—The bending moment M is the same as in the first case, expression (28-11): the torque T is given by expression (28-31) and the equivalent torque T_e may be found using expression (28-14). The shear stress due to T_e must not exceed 5000 to 6000 psi.

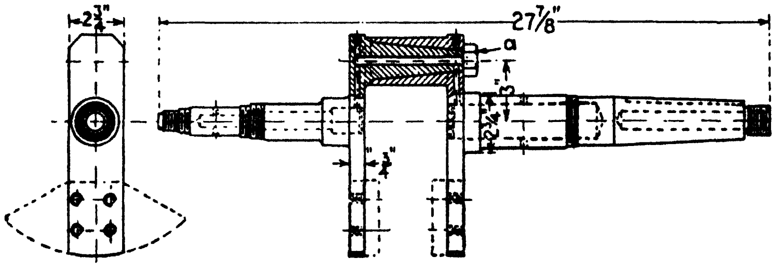


FIG. 28-11.—Crankshaft of a radial aircraft engine.

All allowable stresses given above refer to crankshafts forged of an open-hearth steel. For alloy-steel and cast crankshafts the values of the allowable stresses must be changed on the basis of data given in Table 28-1.

Constructions.—The center crankshaft of stationary engines is usually built in one piece. The crankshafts of radial engines are often made of two pieces to permit the use of a one-piece master connecting rod or cluster. The two halves of the shaft are fitted one into another on a lapped-in conical surface, Fig. 28-11, or sometimes on cylindrical surfaces. Tightening of the nut *a* or in other designs of a through bolt creates the necessary friction, which holds the two halves together; a round key is driven between the two halves; it is more a matter of precaution because all the torque is transmitted through the right half, and the inner part of the crankpin and the left cheek serve only to reduce the bending stresses on the working crank.

28-3. Multithrow Crankshafts.—The problem of determining the stresses in a multithrow crankshaft is rather laborious if done properly and involves certain assumptions in regard to the center line of the shaft. In these computations it is assumed that the crankshaft is correctly aligned and its center line is perfectly straight. When the engine begins to operate, one or several of the bearings will wear faster than the rest, the crankshaft center line will sag at these bearings, and the bending stresses will be increased greatly, but undefinably unless the deflections are measured. Similar conditions arise if the bedplate or crankcase is not rigid.

For these reasons actual designs are very often based on experience without reference to calculations, other than of simple proportion. Another commonly used method is to determine the dimensions of the crankpin, crank webs, and journals as for a single-throw shaft, disregarding the influence of the other cylinders except for the torsional moments. The use of low allowable stresses can take care of the simplifying assumption.

Polar load diagrams, as explained in Sec. 26-5, must be used when designing all but low-speed engine crankshafts, in regard both to strength and to size of bearing areas. The influence of the forces of inertia of both the reciprocating and rotating masses must be taken into account as accurately as possible, especially for high-speed engines.

Finally, it must be kept in mind that a good crankshaft must be rigid. Absence of appreciable deflections helps to obtain uniform bearing pressures and proper lubrication, and reduces the wear of the bearings.

Empirical Formulas.—Crankshafts of marine oil engines must comply with certain empirical rules set up by large insurance

TABLE 28-3.—COEFFICIENT *a* IN AMERICAN BUREAU OF SHIPPING FORMULA

Number of cylinders		S/L ratios							
Four-stroke	Two-stroke	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4
		Spark-ignition engines							
1-2-4	1-2	1.17	1.17	1.17	1.17	1.17	1.17	1.17	1.17
3-5-6	3	1.17	1.17	1.17	1.17	1.19	1.20	1.22	1.24
8	4	1.17	1.19	1.21	1.23	1.25	1.28	1.30	1.32
10-12	5-6	1.18	1.20	1.23	1.25	1.28	1.31	1.33	1.35
		Air-injection diesel engines							
1-2-4	1-2	1.17	1.19	1.22	1.25	1.28	1.31	1.34	1.36
3-6	3	1.19	1.22	1.25	1.28	1.32	1.35	1.38	1.41
8	4	1.20	1.24	1.27	1.30	1.33	1.37	1.40	1.43
12	6	1.22	1.25	1.29	1.32	1.36	1.39	1.42	1.45
16	8	1.25	1.29	1.33	1.36	1.40	1.44	1.47	1.50

companies. These formulas differ one from another, but all give practially identical dimensions; therefore only one set, that of the American Bureau of Shipping, will be given. It also may be mentioned that these rules give a stronger shaft than usual computations would give.

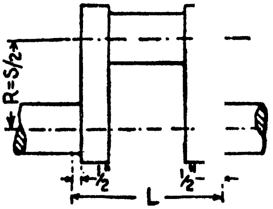


FIG. 28-12.—American Bureau of Shipping designations.

The diameter d of the crankpins and journals is found from the expression

$$d = a\sqrt[3]{D^2pL/S_s} \qquad (28\text{-}32)$$

where a is the coefficient given by Table 28-3,

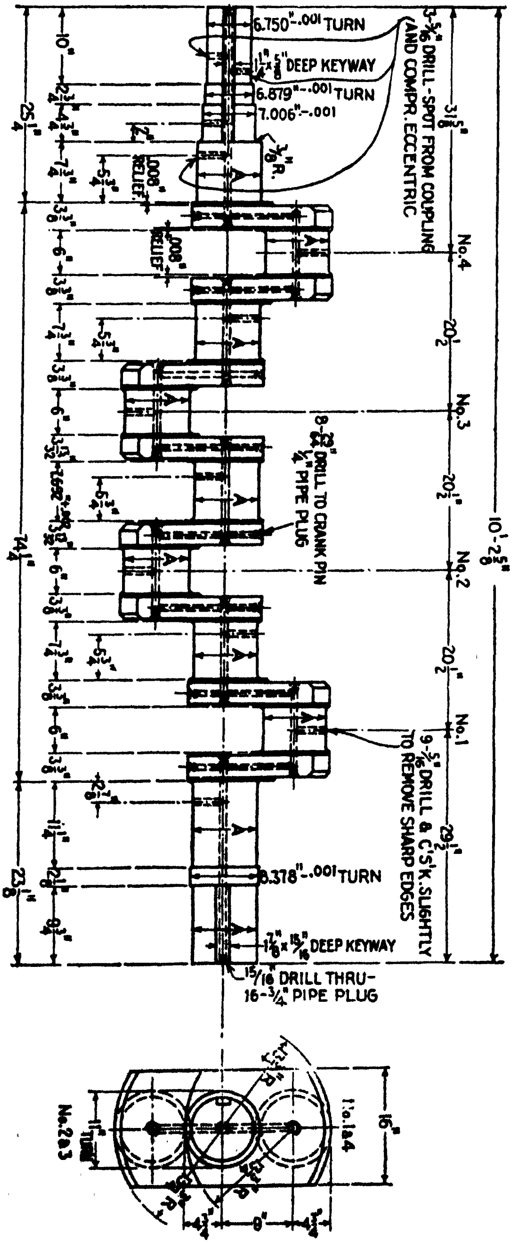
D is the cylinder bore, in.,

p is the maximum gas pressure, psi,

L is the distance over the crank webs, plus 1 in., Fig. 28-12,

S_s is the allowable stress: 7000, psi for cast-steel shafts, 7500 psi for open-hearth steel, and 8000 psi for best grade forged-steel shafts,

S is the engine stroke, in.



FINISH ALL OVER-
ALL FILLETS TO BE 3/8" RADIUS TO GAUGE
TURN AND POLISH ALL DIAMETERS
MARKED "A" TO 8.250-.001

OPEN HEARTH STEEL FORGING-
TENSILE STRENGTH NOT LESS THAN 75,000 LB. PER SQ. IN.-
ELASTIC LIMIT NOT LESS THAN 37,500 LB. PER SQ. IN.
DO NOT LEAVE LESS THAN ⅛" NOR MORE THAN 5/32" OF
METAL FOR CLEANING ON ROUGH TURNED JOURNALS,
AND 3/16" TO ¼" ON CRANK PINS:-
DIMENSIONS SHOWN ARE FOR FINISHED SHAFT

FIG. 28-13.—Shop drawing of a crankshaft for a 13-in. × 18 in. 260 rpm oil engine.

The thickness t and width w of the crank cheeks, also called *webs*, can be selected in much the same way indicated for the single-cylinder shaft, but they must satisfy the conditions:

$$t^2w \geqq 0.4d^3 \tag{28-33}$$

and

$$tw^2 \geqq d^3 \tag{28-34}$$

Torsional Vibration.—Regardless of how the shaft was designed, it should be checked for critical speeds as explained in Sec. 25-7. If the critical speed is too close to the normal engine speed, the shaft diameter must be increased until the shaft is safe from torsional vibration.

Constructions.—Figure 28-13 shows a drawing of a four-throw crankshaft for a 13-in. × 18-in. oil engine. The shaft is drilled for lubricating the crankpins and wristpins. As mentioned before, one of the journals, between cylinders 2 and 3, locates the shaft lengthwise. All junctions of different sections have fillets of $\frac{3}{8}$ in., although latest indications are that in order to be effective the fillets should be equal to one-fifth of the shaft diameter.

Multithrow crankshafts of large engines are built up of several pieces, either shrunk together, Fig. 28-3, or bolted with flanges.

28-4. Side-crank Shaft. *Advantage.*—This type of shaft is used in America for medium-size and large horizontal engines.

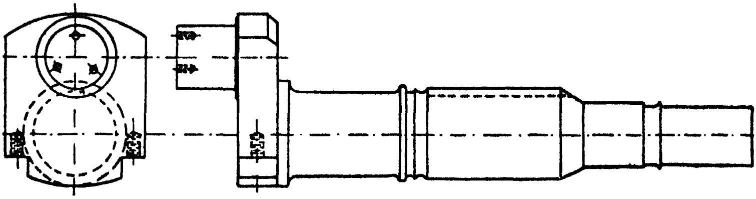


FIG. 28-14.—One-piece side-crank shaft.

Its main advantage is that it requires only two bearings, both in the single-crank or two-crank constructions. With only two bearings there is much less danger of misalignment, which causes most of the shaft failures.

Constructions.—In smaller engines the crankshaft usually is made a one-piece forging with a cast-iron counterweight bolted to the crank, Fig. 28-14. In larger engines the shaft, crank, and pin are made separate pieces and forced together with hydraulic pressure, Fig. 28-15. The crank and counterweight usually are one steel casting. The key k is more of a safety device as the press fit must be sufficient to transmit the torque. The main proportions of the crank are indicated in Fig. 28-15. In Fig. 25-15 was shown a side crank, cast together with the crankpin. The drawback of this construction is that it requires a better, more expensive material because of the crankpin.

Designing.—The crankshaft should be designed or checked along the same lines as the center crankshaft: first, with the crankpin on dead center, where the bending moment is a maximum, next, with the crank at the angle where the torsional moment is a maximum. Finally, the minimum oil-film thickness h_m must be checked. If a new crankshaft is designed, the first step is to determine the approximate values for the diameter and length of the crankpin and the journals.

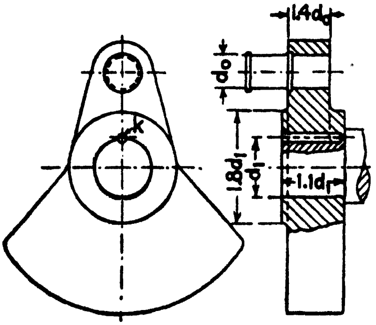


FIG. 28-15.—Three-piece side-crank shaft.

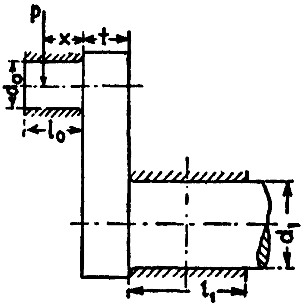


FIG. 28-16.—Side crank.

For the crankpin it is convenient to assume the ratio of $l_0/d_0 = k$, Fig. 28-16, where k varies from 0.8 to 1.1. The bending moment on the pin from an evenly distributed load would be $0.5Pl_0$. Actually the pressure may not be distributed evenly and may give a greater bending moment, up to Pl_0. Assuming as a safe figure the mean value, or $0.75Pl_0 = 0.75Pkd_0$, and equating it to the moment of resistance gives

$$0.75Pkd_0 = 0.0981d_0^3S \tag{28-35}$$

where the allowable stress may be taken as 8000 psi. From equation (28-35)

$$d_0 = \sqrt{0.75Pk/0.0981S} \tag{28-36}$$

After that the bearing pressure P/l_0d_0 should be checked and fall between 1200 and 1600 psi. Finally Zn/p and h_m must be checked.

The thickness t of the crank cheek can be assumed from $0.45d_0$ to $0.75d_0$.

The moment bending the shaft in journal 1 is

$$M = P(0.75l_0 + t + 0.5l_1) \qquad (28\text{-}37)$$

where l_1 can be assumed from $1.5d_0$ to $2d_0$. Equating M to the moment of resistance gives

$$M = 0.0981d_1^3S$$

and

$$d_1 = \sqrt[3]{M/0.0981S} \qquad (28\text{-}38)$$

With these preliminary values of d_0, l_0, d_1, and l_1 the checking of the stresses in the two dangerous crank positions can be started.

Crank on Dead Center.—Figure 28-17 shows a crankshaft with its loads and distances of their application at this crank position.

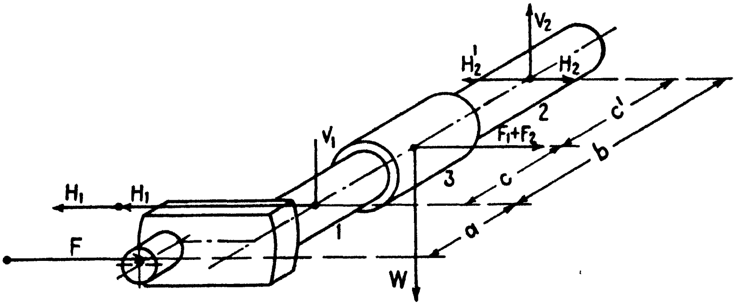


FIG. 28-17.—Forces acting on a crankshaft at dead center.

The gas pressure gives the force F and the two horizontal reactions at the bearings 1 and 2

$$H_1 = F(a + b)/b \qquad \text{and} \qquad H_2 = -Fa/b \qquad (28\text{-}39)$$

The weight of the flywheel W gives two vertical reactions

$$V_1 = Wc'/(c + c') \qquad \text{and} \qquad V_2 = Wc/(c + c') \quad (28\text{-}40)$$

The belt pull $F_1 + F_2$ gives two horizontal reactions

$$H_1' = (F_1+F_2)c'/(c+c') \qquad \text{and} \qquad H_2' = (F_1+F_2)c/(c+c') \quad (28\text{-}41)$$

The distances b, c, and c' must be taken from the preliminary engine layout.

With these forces, reactions, and distances established, the bending moments can now be found and represented graphically, Fig. 28-18, to facilitate both their determination and finding of the resultant moments. In Fig. 28-18 the vertical moments are laid off above the base line 0-2 and the horizontal moments below it.

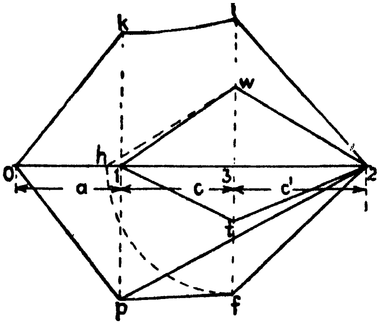


FIG. 28-18.—Diagram of moments.

The horizontal bending moment due to the gas pressure F is a maximum at bearing 1 and is at this point

$$M_1 = Fa \qquad (28\text{-}42)$$

In Fig. 28-18, M_1 is represented by two straight lines, 0-p and p-2.

The other horizontal bending moment due to the belt pull $F_1 + F_2$ is a maximum at point 3, center of the flywheel pulley,

$$M_{3h} = H_1'c = H_2'c' = (F_1 + F_2)cc'/(c + c') \qquad (28\text{-}43)$$

In Fig. 28-18 M_{3h} is represented by two straight lines, 1-t and t-2. Adding the two horizontal moments gives the moment diagram $0pf2$.

The vertical moment due to the flywheel weight W is a maximum at point 3, being

$$M_{3v} = V_1c = V_2c' = Wcc'/(c + c') \qquad (28\text{-}44)$$

The horizontal and vertical moments can be combined graphically. The procedure is illustrated for point 3: swinging an arc with a radius $3f = M_{3h}$ to the line of abscissas gives the point h; the diagonal connecting the points h and w represents the resultant moment, which laid off vertically from point 3 gives point i. Proceeding in the same manner from point 1 to 3 gives curve ki of total bending moments. The straight lines 0-k and i-2 complete the moment diagram, the ordinates of which represent the total bending moment for any point of the shaft.

Crank.—The bending moment on the crank is, Fig. 28-16,

$$M = F(l_0 + t)/2 \qquad (28\text{-}45)$$

as in this case the application of the force F can be considered at the center of the crankpin. The bending stress will be

$$s_1 = M/Z = 3F(l_0 + t)/wt^2 \tag{28-46}$$

In addition there is a direct compression stress

$$s_2 = F/wt \tag{28-47}$$

The total stress on the fibers of the left side of the crank is

$$s = s_1 + s_2 \tag{28-48}$$

and should not exceed 10,000 psi.

Shaft under Flywheel.—The bending moment M_c may be taken from Fig. 28-18, ordinate $\overline{3i}$, and the diameter d may be determined by expression (28-12), using a low allowable stress S_s of 6000 to 7000 psi. The bending moment at point 3 being a maximum, the shaft diameter at this point is also larger than at the journals.

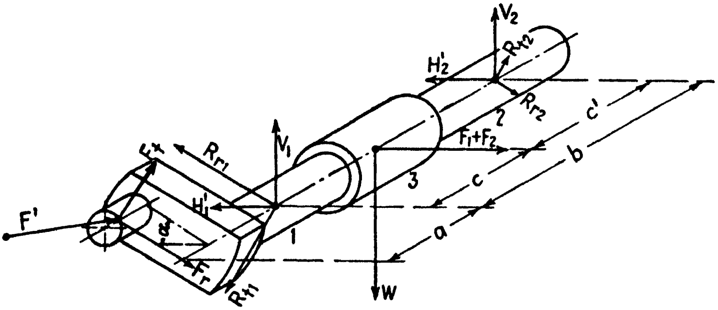


FIG. 28-19.—Forces acting on a crankshaft at angle of maximum torque.

Crank at Angle of Maximum Torque.—The forces acting on the shaft at this position are shown in Fig. 28-19. The angle α is found in the same manner as for the center crank.

The reactions created at the bearings are determined as before: those due to the force F' exerted by the connecting rod on the crankpin can be resolved into radial reactions

$$R_{r1} = F_r(a + b)/b \quad \text{and} \quad R_{r2} = -F_r a/b \tag{28-49}$$

and into tangential reactions

$$R_{t1} = F_t(a + b)/b \quad \text{and} \quad R_{t2} = -F_t a/b \tag{28-50}$$

Together with the reactions H_1', H_2', V_1, and V_2, which are unchanged and given by expressions (28-41) and (28-40), all reactions lie in two planes going through the points 1 and 2 and normal to the center line of the shaft. Thus they may be easily combined into reactions R_{r1} and R_{t1}, also R_{r2} and R_{t2}. The bending moments and combined total bending moments may be found as before, obtaining a moment diagram similar to Fig. 28-18. The torque T acts from the inner end of the crankpin to point 3, and its action upon the resulting stress may be taken into consideration by means of the expression (28-13) or (28-15).

Crank.—The dangerous section of the crank is that where the crank joins the shaft, as this section is subjected to bending due to F_t, bending due to F_r, direct compression by F_r, and to torsion.

The bending stress due to F_t is, with $r - d/2 = r'$,

$$s_1 = 6F_t r'/tw^2 \tag{28-51}$$

That due to F_r is, with the lever arm x, Fig. 28-16,

$$s_2 = 6F_r x/t^2 w \tag{28-52}$$

and the direct compression stress is

$$s_3 = F_r/tw \tag{28-53}$$

The maximum stress will be at the upper left corner of the crank where both bending stresses s_1 and s_2 will be compression and, as for a center crankshaft,

$$s = s_1 + s_2 + s_3 \qquad [28\text{-}26]$$

The torsional moment is

$$T = F_t(l_0 + t)/2 \tag{28-54}$$

and the corresponding stress is

$$s_s = T/Z_p \qquad [28\text{-}28]$$

with the polar section modulus given by expression (28-29).

The total stress s_{max} is found by combining s and s_s by expression (28-13); s_{max} should not exceed 10,000 psi.

Juncture of Crank and Shaft.—The bending moment here is

$$M = F'(l_0/2 + t) \tag{28-55}$$

and the torsional moment is

$$T = F_t r \qquad [28\text{-}31]$$

These moments M and T may be combined using expression (28-14), and the resulting shear stress should not exceed 5000 to 6000 psi.

Shaft under Flywheel.—The bending moment as found in Fig. 28-18 must be combined with the torque, expression (28-31), by means of expression (28-14). The shear stress from this equivalent moment should not exceed 4000 or 5000 psi, which may require an increase of the shaft diameter at this point.

Other Dimensions.—The diameter of the journals should be determined by the same method as under the flywheel and checked for the minimum oil-film thickness h_m. Both journals are usually made of the same diameter. The length of the journals is found from the allowable specific pressure, Table 26-2, and the maximum reaction at the journals.

28-5. Problems.—1. Determine the desirable firing order of a five-cylinder radial four-stroke engine.

2. Determine the desirable firing order of a seven-cylinder radial four-stroke engine.

3. Determine the desirable firing order of a nine-cylinder radial four-stroke engine.

4. Determine the desirable firing order of a seven-cylinder two-stroke radial engine.

5. Find all reactions in a three-bearing center crankshaft of a 10¼-in. × 16-in. horizontal gas engine with the crank on the firing dead center, explosion pressure 340 psi. Referring to Fig. 28-9, assume $a = a' = 11$ in., $l_o = 5$ in., $b = 11$ in., $b' = 12\frac{1}{2}$ in., $t = 3$ in., $w = 7$ in.; the weight of the flywheel $W = 3500$ lb, and $F_1 + F_2 = 700$ lb.

6. Using the data given in problem 5, find the maximum combined stress in the right-hand crank when the crank is at an angle of 35°; the pressure on the piston is 145 psi; the length of the connecting rod is 38 in.

7. For the crankshaft in problems 5 and 6, find the diameter of the crankpin, considering the strength and bearing pressure; check for the oil-film thickness for 250 rpm; assume a suitable lubricating oil.

8. For the crankshaft in problems 5 and 6, find the diameter and length of the journals.

9. For the crankshaft in problems 5 and 6 find the diameter of the shaft under the flywheel.

10. Plot the diagram of bending moments for the side-crank shaft of an 18½-in. × 20-in. horizontal gas engine; the dimensions, as in Figs. 28-16 and 28-17, are: $l_0 = 7$ in., $t = 4\frac{1}{2}$ in., $w = 18$ in., $a = 15$ in., $b = 40$ in.,

$c = 18\frac{1}{2}$ in.; the explosion pressure is 360 psi; the weight of the flywheel $W = 11{,}000$ lb; no belt pull.

11. Plot the diagram of bending moments for the crankshaft of problem 10 when the crank is at 40° past the explosion dead center, the gas pressure is 125 psi, and the length of the connecting rod is $47\frac{1}{2}$ in.

12. Determine the diameter of the crankpin and the diameter and length of the journals for the engine in problems 10 and 11.

13. Check the stresses in the crank of problems 10, 11, and 12.

14. Design the crankpin and the crank for a side-crank shaft for a 20-in. × 24-in. gas engine; the weight of the flywheel is 18,500 lb; the explosion pressure is 350 psi; at maximum torque, when the crank angle is 36°, the gas pressure is 135 psi; the connecting rod length is $4.75r$; select all other dimensions.

15. Design the journals and find the diameter of the shaft under the flywheel for the engine in problem 14.

CHAPTER 29

FLYWHEELS AND GOVERNORS

29-1. Flywheel Action.—*The functions* of a flywheel are:

1. To keep the speed fluctuations within desired limits.
2. With alternators running in parallel, to keep the angular advance or retardation within prescribed limits as compared with a perfectly uniform angular speed.
3. To limit the momentary rise or fall in speed during sudden changes of load.
4. To carry the pistons over the compression pressure when running at low speed.

Flywheel Effect.—Kinetic energy E of a body in foot-pounds is

$$E = Mv^2/2 \tag{29-1}$$

where M is the mass of the body and v is its velocity in fps.

The change of kinetic energy ΔE when the velocity changes from v_1 to v_2 is

$$\Delta E = M(v_2^2 - v_1^2)/2 \tag{29-2}$$

Expressing the velocity by the radius of gyration k in feet and the number of revolutions per minute n, and using for the mass expression W/g, where W is the weight of a flywheel in pounds, give

$$\Delta E = 4W\pi^2k^2(n_2^2 - n_1^2)/(2g \times 3600) \tag{29-3}$$

where n_2 is the maximum and n_1 the minimum speed of the flywheel during one cycle.

Noticing that $n_2^2 - n_1^2 = (n_2 + n_1)(n_2 - n_1)$ and that the average velocity $n = (n_2 + n_1)/2$, designating by m the ratio of average velocity to its fluctuation, or the *coefficient of steadiness*,

$$m = n/(n_2 - n_1) \tag{29-4}$$

and collecting all numerical factors, with $g = 32.2$ fpsps, give

$$\Delta E = Wk^2n^2/2935m \tag{29-5}$$

or

$$Wk^2 = 2935\Delta Em/n^2 \tag{29-6}$$

The quantity $Wk^2 = I_0g$, where I_0 is the polar moment of inertia, is called the *flywheel effect* and is the basis for determining the weight W and size, as expressed by k, of a flywheel. Of the three factors making up the value of Wk^2, the excess energy ΔE must be determined for the particular engine, the coefficient of steadiness of rotation m must be selected according to the service for which the engine is intended, and the engine speed n is given.

29-2. Excess Energy. *Graphical Method.*—The area of the torque diagram divided by its length gives the mean tangential force F_m on the crankpin, Fig. 29-1. If it is assumed that the power taken off the engine shaft is constant, the areas above the mean line represent an excess of energy developed that must be absorbed by the flywheel, and those below indicate a deficiency of energy which must be returned by the flywheel. The weight of the flywheel is based on the maximum excess energy. This may be represented by a single area, as a in Fig. 24-8, or, if the next positive area after the largest positive area is larger than the negative area between them, the maximum excess energy is represented by the sum of the two positive areas minus the area between, as in Fig. 29-1. This means that the flywheel picks up speed while absorbing the energy represented by a and then begins to slow down while returning the energy b. From point 3 to point 4 its speed again increases and, since the area c is larger than b, the speed at point 4 is higher than at point 2 or any other point of the cycle. At point 4 the energy stored in the flywheel reaches its maximum $f = a - b + c$.

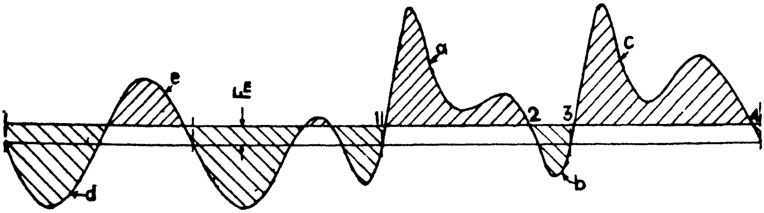


FIG. 29-1.—Torque diagram of a two-cylinder oil engine.

The excess energy ΔE in foot-pounds may be found from the expression

$$\Delta E = fA \times 2\pi rs/12l \tag{29-7}$$

where f is the excess area of the torque diagram, sq in.,
A is the piston area, sq in.,
r is the crank radius, in.,
l is the length of the torque diagram for one revolution, in.,
s is the scale of the ordinates, psi per 1 in.

Analytical Method.—The graphical method requires considerable time and often a more rapid although less accurate method may be preferred. In this method the same expression (29-6) is used, but the excess energy is estimated from former experience with similar engines.

It is logical to expect that the excess energy ΔE absorbed and returned by the flywheel is a function of the work developed by the engine. It has been found that the ratio of ΔE to the engine work referred to one revolution,

$$\Delta E/(33{,}000 \text{ ihp}/n) = \Delta e \tag{29-8}$$

is practically constant for a certain type of engine. Thus from expression (29-8)

$$\Delta E = 33{,}000 \text{ ihp } \Delta e/n \tag{29-9}$$

The values of the constants Δe to be used in expression (29-9) are given in Table 29-1. The lower limits should be taken for low-compression engines, the higher for high-compression engines.

Table 29-1.—Flywheel Constants for Excess Energy

Number of cylinders	Type of engine	Constant, Δe	
		Four-stroke engines	Two-stroke engines
1	Single acting	2.35–2.50	0.95 –1.00
2	Single acting, cranks at 180°	1.40–1.50	0.23 –0.25
2	Single acting, cranks at 360°	0.98–1.05	
2	Single acting, opposed, cranks at 180°	0.98–1.05	0.75 –0.85
3	Single acting, vertical, cranks at 120°	0.55–0.60	0.13 –0.14
4	Single acting, vertical (180° and 90°)	0.15–0.17	0.075–0.085
6	Single acting, vertical (120° and 60°)	0.04–0.05	0.020–0.025
8	Single acting, vertical (90° and 45°)	0.03–0.04	0.005–0.006
1	Double acting, horizontal	1.40–1.50	0.20 –0.22
2	Double acting, tandem or twin	0.19–0.21	0.19 –0.21
4	Double acting, twin tandem, cranks at 90°	0.08–0.09	0.07 –0.08

For compression-ignition oil engines the higher values should be increased by another 20 to 25 per cent, depending upon the compression pressure.

29-3. Uniformity of Rotation.—The value of the coefficient of steadiness m, expression (29-4), or of its reciprocal $1/m$, called *coefficient of uniformity of rotation* and designated as u, depends on the nature of the service for which the engine is intended, *i.e.*, the requirements in regard to uniformity of rotation of the driven machinery and the method of connecting the engine to it.

With the former designations

$$u = 2(n_2 - n_1)/(n_2 + n_1) \qquad (29\text{-}10)$$

With a flexible connection, as a belt drive with long center distance, the coefficient m may be smaller than with a less flexible connection. A paper-making machine requires a more uniform drive, a greater m, than a pump; and an electrical generator a still greater one.

TABLE 29-2.—COEFFICIENT OF STEADINESS m

Driven machinery	Type of drive	m
Hammers, crushers	Belt	5
Excavators, concrete mixers, compressors	Belt	12
Pumps, shears	Belt or flexible coupling	20
Metal and wood-working machines	Belt	30
Textile, flour, and paper mills	Belt	40
Pumps, compressors, and similar machines	Gear transmission	50
Spinning machinery—coarse to fine	Belt	50–65
D-c generator, single	Belt—direct coupled	35–50
D-c generators, parallel	Belt—direct coupled	50–70
A-c generator, single	Belt—direct coupled	60–90
A-c generators, parallel	Direct coupled	100
Boat propeller, up to 100 hp	Direct coupled or by gears	20–30
Boat propeller, over 100 hp	Direct coupled or by gears	40
Automobile engine	Idling	5
	Normal speed	100

Table 29-2 gives minimum values of m for a number of typical drives, the maximum limit, as used in practice, being as much as 25 per cent higher. The computation of the flywheel inertia for parallel operation of a-c generators requires not only fairly high values of m, 100 or over, but also fulfillment of other conditions determined by the electric characteristics of the generators

In a variable-speed engine, such as marine or automobile engines, the flywheel inertia must be sufficient to overcome compression pressures at the lowest speed at which the engine may be operated. Table 29-1 shows that multicylinder engines from six cylinder up, in the two-stroke, and over eight cylinders, in the four-stroke type, practically do not require any flywheel. Radial airplane engines also do not need a flywheel, the counterbalanced crankshaft and propeller providing a sufficient moment of inertia even at low speeds.

29-4. Flywheel Design. *Main Dimensions.*—In finding the weight of a flywheel, usually the inertia of the rim only is considered, neglecting the small influence of the arms and hub. First, the radius of gyration k in the expression Wk^2 is determined by assuming a proper rim velocity $v = \pi D n$. This should not exceed 5000 fpm for cast-iron wheels of engines under 100 hp, and 6000 fpm for larger engines. With a special rim design which prevents blowholes in the rim, the rim velocity may be increased to 7000 fpm and up to 10,000 fpm for cast-steel wheels. In automobile engines, rim speeds of cast-iron flywheels reach 10,000 fpm, those of semisteel wheels 15,000 fpm, and those of steel wheels 20,000 fpm.

The relation between k and outside diameter D of the rim is

$$k^2 = 0.125[D^2 + (D - 2a)^2] \qquad (29\text{-}11)$$

If the rim thickness a is small as compared with D, it may be neglected, giving $k = 0.5D$.

After the weight is found from expression (29-6), the rim thickness a and width b in inches may be computed from the expression of the volume times the specific weight

$$W = 24\pi kabw \qquad (29\text{-}12)$$

where for cast iron $w = 0.26$ and for steel $w = 0.28$ lb/cu in. The ratio a/b is selected between 0.5 and 1.5. If the flywheel is to be used also as belt pulley, the width b must be made 1 or 2 in. larger than the belt width. The outside diameter is $D = 2k + a$. The hub diameter may be made equal to two shaft diameters, and the hub length should be 2 to $2\frac{1}{2}$ shaft diameters.

Construction.—Flywheels up to 8 ft in diameter are cast solid; above this size they are made in halves. A split hub, Fig. 29-2, by clamping the shaft prevents the key from working loose. The hub is bored only 0.001 in. smaller than the shaft, and a steel wedge driven into the hub slot helps to put the wheel on the shaft. The diameter of the clamping bolt is made about one-sixth of the shaft diameter. Some designers split the hub from both sides, as shown by dotted lines, Fig. 29-2.

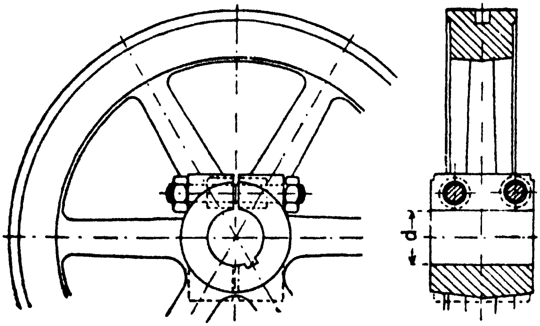


FIG. 29-2.—Flywheel with a split hub.

Flywheels made in two halves should be parted on an arm and not between the arms. The halves are connected by bolts through the hub and near the rim with additional shrink links, Fig. 29-3, or shrink anchors, Fig. 29-4. An anchor connection with wedge keys, Fig. 29-5, may be used to avoid the troubles encountered with shrink fits, especially if the flywheel has to be taken off eventually.

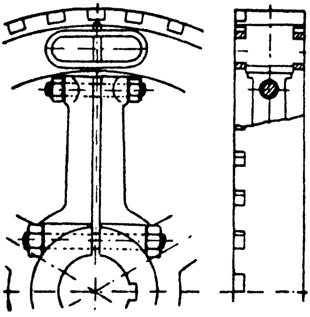


FIG. 29-3.—Flywheel cast in two halves.

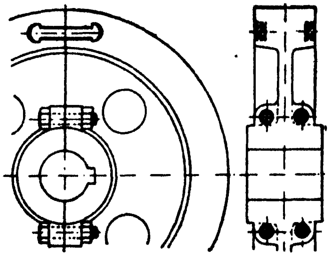


FIG. 29-4.—Split-disk flywheel.

Marine-engine flywheels are often made with a web fastened by six fitted bolts to a separate steel-forged hub, resembling, in this respect, automobile flywheels. Figure 29-6 shows three typical automobile flywheels, *a* for a six-cylinder and *b* and *c* for eight-cylinder engines.

The automobile flywheel rims have teeth for the starter engagement. Stationary- and marine-engine flywheel rims also need teeth in larger sizes and barring holes in smaller sizes to turn the crankshaft to starting position and for checking the timing, Figs. 29-2 and 29-3.

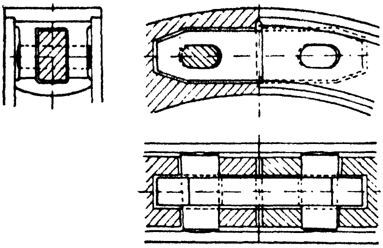


FIG. 29-5.—Anchor with wedge keys for flywheel rim.

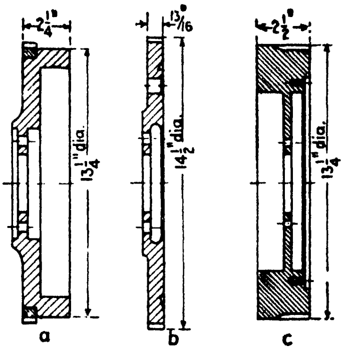


FIG. 29-6.—Types of automobile flywheels.

Balancing.—Flywheels of automobile and other high-speed engines are machined all over and additionally balanced on special machines to ensure freedom from vibration. Flywheels of low-speed engines with cast arms should be balanced at least statically. This is done usually by drilling holes in the heavier part of the rim and driving wooden plugs into these holes. After the wheel is painted, the holes are hardly noticeable.

Stress in Rim.—The centrifugal force exerts a uniformly distributed radial load over the entire circumference of the rim. The general expression for the centrifugal force is

$$C = Mv^2/R = Wv^2/gr \tag{29-13}$$

Substituting in this expression the weight of the rim per linear inch $W/24\pi k$, for the velocity $v = 2\pi kn/60$ fps, $g = 32.2$ fpsps, and $r = k$ ft, gives

$$C = 0.00000452Wn^2 \tag{29-14}$$

and the component of the total force normal to any diameter is

$$C_n = C \times 2k \times 12 = 0.0001084Wn^2k \text{ lb} \tag{29-15}$$

This force is applied to the rim sections at the ends of the diameter, each equal to ab sq in. producing a tensile stress

$$s = 0.0000542Wn^2k/ab \tag{29-16}$$

or substituting for W its expression (29-12)

$$s = 0.0041n^2k^2w \tag{29-17}$$

Arms.—The standard number of arms is six, and only very wide flywheels serving as belt pulleys are made with two rows or 12 arms. Large oil engines sometimes have flywheels with webs instead of arms, Fig. 29-4. The holes are made to facilitate handling.

The arms usually have an elliptical section with the major axis twice the minor, and taper down toward the rim from 10 to 25 per cent. They should be strong enough to withstand the maximum bending moment coming from the piston when starting the engine under load. The torque diagrams show that this moment is approximately Apr lb-in., where A is the piston area in square inches, p is the maximum gas pressure, psi, and r is the crank radius, in. For safety's sake this moment is considered distributed equally among only one-half at the arms so that if i is the number of arms, the moment on one arm becomes, using $A = \pi D^2/4$ and $r = l/2$, D and l being engine bore and stroke,

$$M = \pi D^2pl/4i \tag{29-18}$$

The section modulus of an ellipse is $Z = \pi ab^2/32$, or $\pi b^3/64$, when the minor axis is one-half the major. Thus the stress is

$$s = M/Z = 16D^2lp/ib^3 \tag{29-19}$$

which gives for the major axis

$$b = 2.52\sqrt[3]{D^2lp/is} \tag{29-20}$$

Under the conditions mentioned above, a stress of $s = 2000$ psi can be allowed. The value b determined by (29-20) would be at the center of the shaft. Laying out the arm with a taper as indicated will give the values b_1 at the hub and b_2 at the rim and the minor axes $a_1 = b_1/2$ and $a_2 = b_2/2$.

After this the arms should be checked for the stress due to the centrifugal force when the engine is running at maximum speed. The total centrifugal force is equal to $2C$ as found from formula

(29-15), and the force acting upon each arm is $2C_n/i$. The tensile stress s_1, due to the centrifugal force of the rim, is equal to $2C_n/i$ divided by the area of the ellipse, $A = 0.7854a_1b_1$. To this may be added a stress s_2 due to the weight of the corresponding part of the rim. If the flywheel is used as a belt pulley, a third stress may be computed, due to the bending moment of the tangential force $(F_1 - F_2)$ of the belt drive, which is, again assuming that only one-half of the arms carry the load,

$$M_b = (F_1 - F_2)(D - d)/i \tag{29-21}$$

where D is the flywheel and d the hub diameters. Dividing this moment by the section modulus $Z = 0.1a_1b_1^2$ gives the bending stress s_3 at the hub. The total stress $s = s_1 + s_2 + s_3$. If this total stress exceeds 3000 psi, the arm section should be increased.

Stress Due to Acceleration.—When a large load is thrown on an engine suddenly, the engine begins to slow down, but the flywheel tends to maintain the normal engine speed and the inertia of its rim may throw a considerable bending stress into the arms. A quick stopping of the engine may be considered as a limit case.

To find the corresponding stress, first the time of stopping t in seconds must be known or assumed. The force F necessary to stop the wheel in that time can be presented as

$$F = Ma = Wa/g \tag{29-22}$$

where the negative acceleration

$$a = v/t \tag{29-23}$$

The force F is applied at the center of gravity of the rim and will bend the arms at the hub with a moment

$$M_b = F(12k - d/2) \text{ lb-in.}$$

Example 29-1.—Find the stress in the arms of a flywheel which is stopped in three revolutions from a speed of 260 rpm. The radius of gyration $k = 2.75$ ft, the moment of inertia of the rim $Wk^2 = 30{,}500$ lb-ft^2, the hub diameter is 14 in., and the arm section is an ellipse with axes of 3.5 and 7 in.

The velocity of the center of gravity

$$v = 2\pi kn/60 = 2\pi \times 2.75 \times 260/60 = 75 \text{ fps}$$

The time of deceleration is the distance traveled divided by the average velocity

$$t = 2\pi 2.75 \times 3/[(75 + 0)/2] = 1.383 \text{ sec}$$

which gives

$$a = 75/1.383 = 54.2 \text{ fpsps}$$

The weight of the rim is

$$W = 30{,}500/(2.75)^2 = 4033 \text{ lb}$$

and the force necessary to stop the wheel in three revolutions

$$F = 4033 \times 54.2/32.2 = 6810 \text{ lb}$$

The moment

$$M_b = 6810(12 \times 2.75 - 14/2) = 177{,}000 \text{ lb-in.}$$

The section modulus of the arm section

$$Z = 0.1 \times 3.5 \times 7^2 = 17.15 \text{ in.}^3$$

Thus, assuming that three of the six arms carry the load

$$s = 177{,}000/(17.15 \times 3) = 3{,}440 \text{ psi}$$

The stress appears rather high so that it would be advisable to increase the size of the arm section.

29-5. Governor Characteristics.—The function of a governor is to maintain a desired engine speed irrespective of the changes in the load. A governor accomplishes this by controlling the amount of fuel admitted. The agent that actuates every governor is the centrifugal force of a pair of rotating flyballs.

All governors may be divided into two groups: (1) direct-action governors and (2) relay-type governors.

In direct-action governors the work of controlling the mechanism which admits the fuel is done by the flyballs themselves. In relay-type governors the flyballs actuate a relay, in most cases a slide valve, which admits oil under pressure to the one or the other end of a power cylinder with a piston in it; the oil moves the piston which does the work necessary to change the fuel admission. These governors also are called *hydraulic* or *fluid* governors.

The advantages of a direct-action governor, as compared with a relay-type one, are: (*a*) simplicity, (*b*) smaller size, and (*c*) lower cost.

The disadvantages of a direct-action governor are: (*a*) inability to maintain a close speed regulation, particularly a constant speed of the engine, and (*b*) limited power.

If the load conditions demand an exceptionally good speed control. the complication and expense of a relay-type governor are justified. With it almost any desired degree of speed control may be obtained, and any number of compensating features may be added to the control mechanism. A relay-type governor can be made to operate even on a 0.01 per cent speed change. The relay-type governor is a machine in itself and, as such, is built by only a few special manufacturers. However, with the wide introduction of electric clocks most of the utilities were forced to use this type of governor on their prime movers, whether of the steam, hydraulic, or internal-combustion-engine variety. In the following only the direct-action governors will be discussed because they are still widely used.

Governor Characteristics.—The important characteristics are:

1. Closeness of regulation.
2. Sensitivity.
3. Stability.
4. Power.

Closeness of regulation is expressed as the ratio p of the difference between n_0, the speed at no-load condition, and n_1, that at full load, to the average speed. The latter can be set equal to $(n_0 + n_1)/2$ and hence

$$p = 2(n_0 - n_1)/(n_0 + n_1) \tag{29-24}$$

Speed droop, designated by p', is a term more often used. It refers the speed fluctuation to the full-load speed n_1, usually called *rated speed,*

$$p' = (n_0 - n_1)/n_1 \tag{29-25}$$

Evidently $p' > p$, although, practically, they can be considered as equal. Speed droop usually is expressed in per cent. For engines driving electric generators in parallel and equipped with direct-action governors a speed droop of about 4 per cent helps the governors to divide the load between the units. Where the load conditions require close regulation, a speed droop of only 2 per cent can be obtained without undue complications. Engines driving alternators operated in parallel or when a standard frequency is required must have some means for speed adjustment, such as changing slightly the tension of the governor spring or the fulcrum of a floating lever.

Closeness of regulation and speed droop of an engine in operation are determined by running it from no load to full load or a desired overload, as the case may be, and measuring the engine speed at various loads. Figure 29-7 presents graphically the results of such a test.

Example 29-2.—Find the closeness of regulation and speed droop for the engine represented on Fig. 29-7.

The rated speed is 504 rpm and the closeness of regulation is, by expression (29-24),

$$p = 2(520 - 504)/(520 + 504) = 0.0312$$

or about 3.1 per cent. The speed droop is, by expression (29-25),

$$p' = (520 - 504)/504 = 0.0318$$

or practically 3.2 per cent.

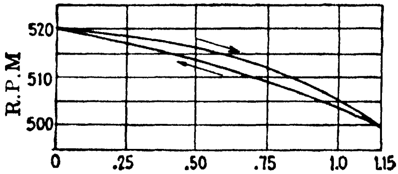


FIG. 29-7.—Sensitivity of governing.

Sensitivity.—If an engine is running and its governor is in equilibrium, it requires a certain change of the engine speed, when the load changes, before the governor begins to act and to adjust the fuel delivery to correspond to the new load. This lag of governor action is caused by friction and lost motion in the governing mechanism and is called the *sensitivity* of the governor. Sensitivity is determined by testing an engine with an increasing and then a decreasing load, as illustrated by Fig. 29-7. Sensitivity q is expressed as the speed difference at a certain load over the average speed at this load

$$q = 2(n' - n'')/(n' + n'') \qquad (29\text{-}26)$$

Example 29-3.—Find the sensitivity of the governor whose characteristics are represented by the curves of Fig. 29-7.

The greatest speed difference is at 0.75 of the full load, with $n' = 509$ and $n'' = 512$ rpm. Therefore, by expression (29-26)

$$q = 2(512 - 509)/(512 + 509) = 0.00588$$

or about 0.6 per cent.

It should be noted that, while it is desirable to have a governor with a small internal friction, a low value of q, q should not be smaller than the uniformity of rotation u, expression (29-10), which depends upon the effect of the flywheel. Since the value of u for a given engine expresses speed fluctuations during each cycle, if $q < u$, the governor will be continuously fluttering and will cause wear of various pivots.

Stability is the ability of the governor to assume and hold a definite position for any given speed and to return to this position of equilibrium after it has been displaced from it. Stability is obtained by using a spring whose tension increases faster than the centrifugal force of the governor weights. A stable governor does not have any appreciable movement because of the reactions from the fuel-control mechanism.

If a governor lacks stability, it may change its position without any appreciable change of the engine load, will *hunt* or *race*, and cause the engine speed to fluctuate.

Hunting occurs because of the lag in action of the control mechanism. It may be caused by insufficient sensitivity or by lack of power. In either case the engine will slow down or speed up too much before a corrective regulation of the fuel is made. When the controls begin to move, an overcorrecting of the engine speed in the other direction will result. After that the governor will start to act in the opposite direction.

Power.—A direct-action governor must furnish the energy to do the work of regulating the admission of the fuel or fuel and air. All of the requirements of a good speed control depend upon this energy. The power of a governor can be increased by increasing the centrifugal force of its weights, as will be shown later.

29-6. Governor Types.—Direct-action governors used with internal-combustion engines are of the centrifugal spring-loaded type, some having a horizontal, the majority a vertical axis of rotation. Whether a horizontal or a vertical governor is to be used depends on the available space and the location of the governor, the kind of movement which is desired, and the type of drive suitable under the given conditions. Two kinds of movement are obtainable—a linear displacement along the axis of rotation and an angular movement in the plane of rotation. The first kind is more often used because it is equally adaptable for both gas and oil engines, whereas the angular movement is readily used only for varying the fuel-pump delivery of oil engines.

Drives.—The problem of the governor drive depends to a great extent upon its location. Only seldom is the rotating mass of the governor fastened directly to the crankshaft or camshaft. Usually the governor shaft is connected to one of these shafts either by a pair of gears, spur, bevel, or helical, or by a chain or belt. A gear drive is preferable; a chain or belt drive is less positive, although convenient for selecting and locating the governor. Both the belt and chain may break or slip off, causing the engine to run away and be wrecked unless it is provided with an additional overspeed governor.

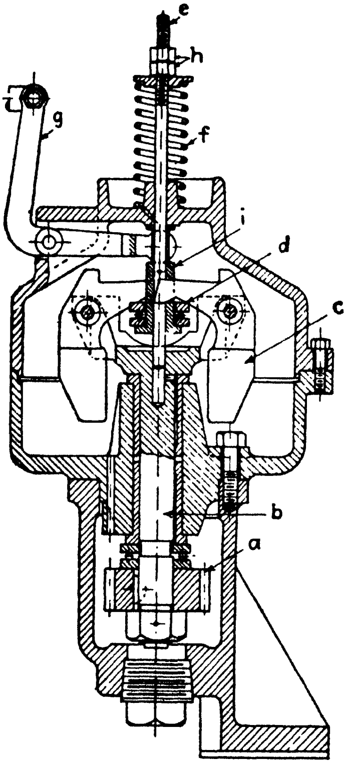


FIG. 29-8.—Governor of a Western-Enterprise gas engine.

Constructions.—A typical gas-engine governor with a vertical axis is shown in Fig. 29-8. The helical gear *a* is driven by the cam gear by means of two intermediate gears and drives the yoke shaft *b* with the four governor weights *c*. Under the action of the centrifugal force the weights *c* tend to move away from the axis of rotation and push down with their fingers the ball thrust bearing *d* and, with it, the sleeve *i* and spindle *e*, compressing the spring *f* and acting upon the bell crank *g*. Crank *g* acts upon the throttle valve which admits the charge. By changing the spring tension by means of nuts *h*, *h*, the speed of the engine can be changed without stopping it.

The same governor could be used for an oil engine, acting either upon a suction release valve of the fuel pump, as shown in Fig. 9-3, or upon the mechanism which controls the lift of mechanically operated fuel injectors.

Figure 29-9 shows a combined hand and governor control for a variable-speed compression-ignition oil engine. The speed is set by the manually operated lever a or the foot pedal b; the tension spring s has a tendency to move the pump rack r to the stop, or no-fuel, position. Once a certain speed is set, the flyballs maintain it by means of the lever c and take care of the load variations.

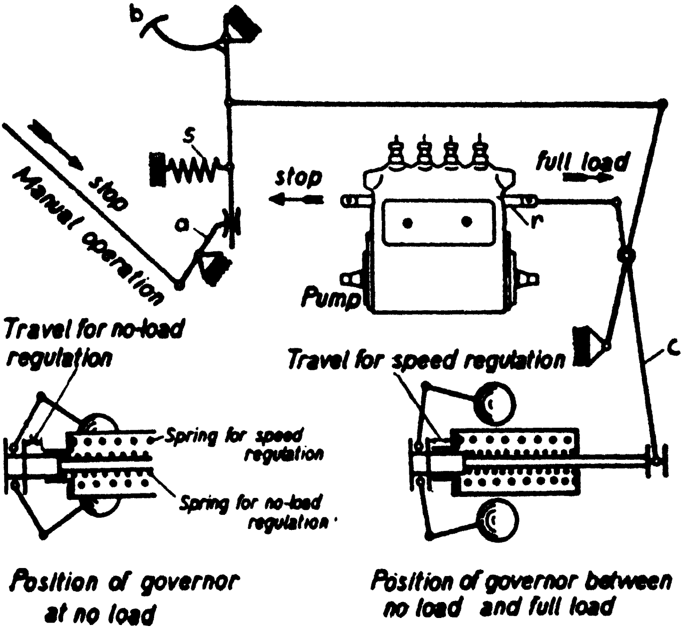

b
s
stop
Manual operation
a
stop
full load
r
Pump
Travel for no-load regulation
Travel for speed regulation
c
Spring for speed regulation
Spring for no-load regulation
Position of governor at no load
Position of governor between no load and full load

FIG. 29-9.—Method of governing which provides regulation for speed and no-load conditions.

Principle of Action.—The action of a flyball governor is based on the centrifugal force

$$C = Wv^2/gr \qquad [29\text{-}13]$$

where in this case W is the combined weight of the governor weights c, Fig. 29-8; v is the velocity; and r is the distance in feet of the center of gravity of c from the axis of rotation. Substituting for $v = 2\pi rn/60$, fps, and $g = 32.2$, gives the force in lb

$$C = 0.000341rn^2W \qquad (29\text{-}27)$$

Expression (29-27) shows that the power of a governor increases in direct proportion to r and W, which means in proportion to its size and as the second power of the speed n. This explains why most governors are geared to run two to three times faster than the engine itself.

Figure 29-10 shows the scheme of a flyball governor: The balls b, b are shown in their inner position, *i.e.*, lowest speed and maximum fuel supply; r_1 is the corresponding radius of gyration in expression (29-27); the dotted lines show the balls in their outer or no-fuel position and at highest speed; r_2 is the corresponding radius of gyration. The reaction F_s of the spring when the balls are in a position of equilibrium is

$$F_s = Cl_1/l_2 \qquad (29\text{-}28)$$

where l_1 and l_2 are the lever arms of the ball crank, Fig. 29-10; i is the collar acting as a spring seat.

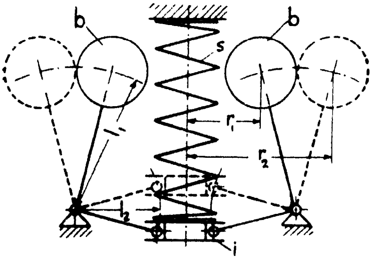


FIG. 29-10.—Principle of action of a flyball governor.

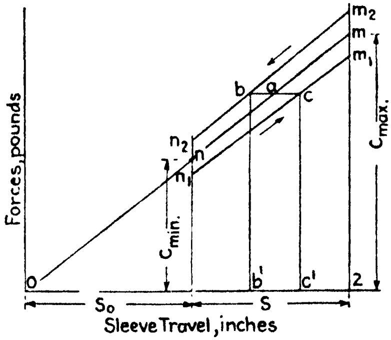


FIG. 29-11.—Relation between forces and sleeve travel.

Governor Design.—If a governor is designed with an axial motion of the sleeve, as in Fig. 29-8 or 29-10, a preliminary sketch is made, conforming to the allotted space. The number of weights c is decided upon, preferably four to obtain better balance, and their size and weights determined. Next, the minimum and maximum distances r_1 and r_2 at the two limit positions and the corresponding travel of the sleeve i are found. Having decided upon the desirable speed droop, expression (29-24), and the speed reduction between the drive shaft and the governor, the idling speed n_0 and full-load speed n_1 are computed. With these data the centrifugal forces C_{max} and C_{min} are figured for the two limit positions by formula (29-27). The determination of the spring can be made graphically, Fig. 29-11: The ordinates represent centrifugal forces, the abscissas the travel of the governor sleeve, s being the total travel between the two limit positions. The centrifugal forces are balanced by, and therefore are equal to, the spring-compression forces. Connecting points m and n by a straight line and continuing it to point o gives the total compression of the spring $s_0 + s$. The influence of friction can be represented by the lines n_1-m_1 running below the line n-m when the load is increased and n_2-m_2 above it when the load decreases. For a position of equilibrium a, the points of intersection b and c with a horizontal line give the position b' or c', as the case may be, of the sleeve, which in turn enables one to compute the corresponding speed n_b or n_c and check the sensitivity by expression (29-26). Usually after the preliminary design it will be found that certain dimensions should be changed, and the computations are repeated.

Requirements to be met in a good governor design are:

1. Enough power, *i.e.*, a large centrifugal force.
2. Use of ball bearings to reduce friction of rotation.
3. Use of bushings and hardened pins in other joints.
4. Well-planned lubrication, *i.e.*, proper supply and systematic disposal of the excess oil.
5. Provision for changing the spring tension without stopping the engine for adjusting the speed.

29-7. Problems.—1. Find the necessary flywheel effect for a four-cylinder 10-in. × 14-in. × 300-rpm four-stroke compression-ignition oil engine; the desired coefficient of steadiness $m = 45$. The torque diagram has an excess area of 3.6 sq in.; its length is 6 in., and the scale of ordinates is 10 psi per inch.

2. Find the necessary flywheel effect for a single-acting vertical three-cylinder two-stroke compression-ignition 120-hp oil engine, operating at 325 rpm, driving a d-c generator by means of a belt.

3. Find the main dimensions of the flywheel in problem 2.

4. Find the necessary flywheel effect for a two-stroke four-cylinder natural-gas engine, rated 220 hp at 257 rpm and driving an a-c generator by means of a belt.

5. Find the main dimensions of the flywheel in problem 4.

6. Find the necessary flywheel effect for a double-acting horizontal two-cylinder four-stroke 2000-bhp blast-furnace gas engine operating at 180 rpm and driving a direct-connected a-c generator in parallel.

7. Find the main dimensions of the flywheel in problem 6.

8. Find the stresses in the rim and the size of arms to be used in the flywheel of problem 3.

9. Compute the stress in the arms of the flywheel in problem 4 when the engine load is suddenly increased from full load to 50 per cent overload and the engine slows down to 200 rpm after 10 revolutions.

10. Design a flywheel for a marine four-cylinder gasoline engine; rated power 60 hp at normal speed of 750 rpm; must slow down to an idling speed of 300 rpm. Assume uniformity of rotation and any other needed data.

11. Design a flywheel for a marine six-cylinder four-stroke compression-ignition oil engine. The rated power is 800 hp at 280 rpm; the engine must be able to slow down to an idling speed of 95 rpm. Assume that the propeller load is proportional to the cube of rpm.

12. A flywheel for a two-cylinder vertical four-stroke 60-hp gas engine with cranks at 180° has a rim weighing 4800 lb; the engine has a $9\frac{1}{4}$-in. bore, 14-in. stroke, runs at 320 rpm; the radius of the center of gravity of the rim section is 2.04 ft. Find the uniformity of rotation of the engine with this flywheel.

13. Determine permissible variations of engine speed between full and no load for the engine in problem 12.

14. Give suitable values for the speed droop of the governor and its sensitivity for the engine of problem 1. Draw a diagram along the lines of Fig. 29-7.

15. Work problem 14 for the engine of problem 2.

16. Work problem 14 for the engine of problem 4.

17. Give suitable values for the closeness of regulation, speed droop, and sensitivity of the governor for the engine of problem 12. Draw a diagram along the lines of Fig. 29-7.

18. Find the flyball weights and all other dimensions for a governor for the engine of problems 4 and 16, using a governor similar to Fig. 29-8 and assuming that at the maximum load, 15 per cent over the full load, the horizontal pull of the upper end of the bell-crank lever b must be 10 lb, its travel from maximum load to no-load must be $\frac{3}{4}$ in.

CHAPTER 30

STARTING—REVERSING

30-1. Starting Methods.—Internal-combustion engines are started by:

1. Hand,
2. An electric motor,
3. An auxiliary gasoline engine,
4. Compressed air,
5. An explosive cartridge, and
6. An inertia starter.

Hand starting can be used only for small engines, having a bore not over 4½ in. and not more than four cylinders.

Electric starting is used extensively for automobile and other automotive engines and will be discussed in more detail later.

Auxiliary Engine.—The main engine is cranked by a small 3- to 5-hp gasoline or natural-gas engine, which operates a Bendix drive similar to the one used on automotive engines. The drive is connected to the main engine by V-belts or gears. The auxiliary engine is connected to the V-belt or gear drive by means of a friction clutch and is started by hand cranking. When it attains the proper speed, the clutch is let in and the main engine is turned over until it starts to fire. When it gets under way, the peripheral speed of the flywheel turns the Bendix screw and disengages the drive from the flywheel. After this the auxiliary engine is shut down. This method is used for engines with a bore up to 8 in.

Some compression-ignition oil engines have the starting gasoline engine incorporated in the main engine, Fig. 30-1. Turning the crank 1 opens valve 3 leading to the auxiliary chamber 4. This added space reduces the compression ratio to that of a gasoline engine. As the crank is turned, the butterfly valve 6 permits the drawing of an air-gasoline mixture, from a carburetor attached to flange 8, through the intake valve 9. A high-tension magneto supplying the current to the spark plug 5 is also engaged at the same time. When the engine has made about 700 revolutions, rod 2 automatically releases the shaft turned by the crank 1. This cuts out the auxiliary chamber and magneto and turns the double butterfly valve, shutting off the flow of the gasoline-air mixture, and opens wide the butterfly valve 7 admitting air. The engine begins to operate as a compression-ignition oil engine.

Air starting is a very widely used method and will be discussed in a separate section.

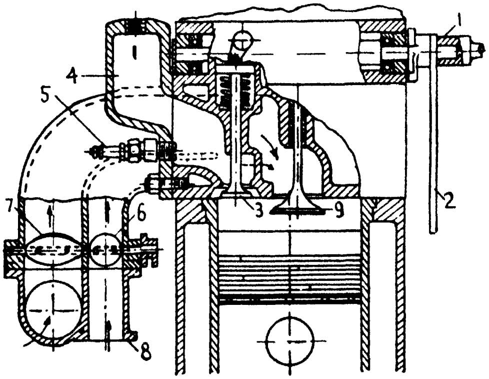


FIG. 30-1.—Gasoline starter of a compression-ignition oil engine.

Cartridge Starter.—A cartridge engine starter consists of two separate devices, the breech and the starter proper. In the breech, a cartridge ignited by an electric current produces high-pressure combustion gases. These gases are admitted through a pipe connection to a piston which, by means of a series of helically splined shafts, first engages the starter jaw with the clutch jaw fastened to the end of the engine crankshaft. When the two jaws are engaged, the continuing motion of the piston produces a rotary motion of the jaws and turns the engine crankshaft, thus starting the engine. When the piston reaches its extreme position, the gases are released through an exhaust valve, the piston is returned to its initial position by a spring, and the starter jaw is disengaged from the crankshaft jaw.

Inertia Starter.—Small aircraft engines are often started by turning the propeller by hand while the pilot is at the controls. This method, because it requires human proficiency, is dangerous; many accidents have occurred when the engine turned backward, after the spark set up before the piston had passed the dead center. In order to increase safety and to enable the pilot to start his engine without outside help, and particularly for starting larger engines, several types of starters have been designed, using compressed air or electric current from a battery.

One of the successful starters is the hand-operated inertia starter of the Eclipse Machine Company. The power for turning the engine is accumulated in a flywheel which is gradually brought up to a very high speed by means of a hand crank. The drive from the hand crank is transmitted through bevel gears and a train of spur gears with internal gears to the flywheel pinion. Through this gear train a flywheel speed of 12,000 rpm may be reached.

The stored energy is returned from the flywheel to the driving barrel through a torque overload release, which is a multidisk clutch with adjustable spring pressure. The engagement with the engine crankshaft clutch jaw is effected by an arrangement similar to the Bendix drive. As soon as the engine starts and its speed exceeds the speed of the starter, the starter jaw clutch is automatically disengaged.

Inertia starters for engines over 200 hp are equipped with electric motors driven from 12- or 24-volt storage batteries.

30-2. Electric Starting.—Automotive, diesel-locomotive, and many marine and industrial engines are started by electric motors. The current is supplied by a storage battery, usually of 6 or 12 volts with a 110 to 150 amp-hr capacity. Marine and locomotive compression-ignition oil engines use 32-, 64-, and 112-volt batteries with a capacity from 100 to 400 amp-hr.

Automobile Starters.—The electric motor is connected to the engine by means of spur gears. Most automobile engines use the Bendix or a similar automatic pinion-shift mechanism. This mechanism consists of a shaft fastened to the motor shaft by means of a flexible coupling. The pinion shaft is a screw with a steep rectangular thread on which the pinion can turn freely and thus move axially approximately 1½ in. When the starter pedal is depressed, it closes a switch between the battery and the motor whose armature begins to spin. The inertia of the pinion assisted by a special counterweight causes the pinion to lag from the speed of the motor shaft; this turns the pinion on the screw, moves it away from the motor, and meshes it with the teeth cut in the rim of the engine flywheel, which it begins to turn. When, after a few turns, the engine starts to fire, the peripheral speed of the flywheel at once exceeds that of the pinion, and the latter is pushed back on its threaded shaft and disengaged from the flywheel gear.

The current required to start an automobile engine is quite large, over 500 amp, as shown in Fig. 30-2, and therefore is not put through the ammeter. The resistance to turning the engine increases with the viscosity of the lubricating oil, and the latter increases very rapidly with the decrease of temperature. Also it is considerably more difficult to obtain the proper gasoline-air mixture at a low atmospheric temperature. Therefore, in cold weather, the current used is much larger and the number of turns of the engine before firing occurs is much greater, resulting in a greater discharge of the battery.

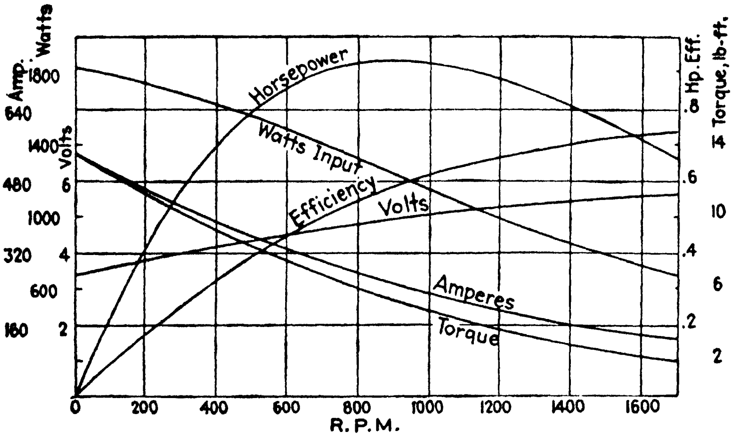


FIG. 30-2.—Typical characteristics of an automobile starting motor.

The battery is recharged by an electric generator of the constant-voltage type which operates when the engine is running. To avoid discharging the battery when the engine is still or running slowly, the generator is equipped with an automatic cutout. If an automobile generator is operated at a greatly varying speed, its efficiency is rather low, varying as maximum values from 30 to 60 per cent. In conjunction with the fairly low efficiency of the starting motor, Fig. 30-2, and the battery losses, the energy which must be supplied by the automobile engine for each starting operation is rather high and affects the gasoline consumption of a car to the extent of 3 to 10 per cent.

Charging characteristics of a 150-amp-hr 32-cell 64-volt battery for cranking a locomotive engine are shown on Fig. 30-3. The battery is charged by an auxiliary generator equipped with a device, which gradually reduces the ampere input, after the battery has reached its full voltage, which is limited by another device. This takes place when the battery is about 80 per cent charged, and after that the ampere input gradually tapers down to about 5 per cent.

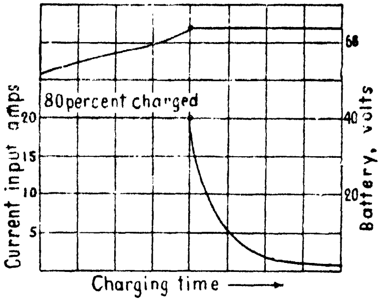


FIG. 30-3.—Charging diagram of a storage battery.

30-3. Air Starting.—Most gas and oil engines, stationary and marine, are started by compressed air.

Starting Air.—The usual air pressure is 150 to 300 psi. Air-injection diesel engines equipped with high-pressure air compressors use air pressures from 500 to 750 psi.

The volume of the air tank necessary to start an engine may be taken as 15 to 25 times the total piston displacement for a small engine, decreasing to about 7 to 10 times the piston displacement for large engines.

Starting valves usually are located in the cylinder heads and operated either directly by cams and levers, in the same way as other valves, or indirectly by compressed air.

The air-admission valve opens 3 to 8° after top center and closes in gas engines 65 to 85° after it, giving an admission of 30 to 45 per cent. In compression-ignition oil engines more air is required and the air-starter valve closes about 55 to 45° before bottom dead center in four-stroke engines and 85 to 75° before bottom center in two-stroke engines. To facilitate the turning over of an engine, in some engines the compression is released during starting by slightly opening the exhaust valve near the middle of the stroke. In stationary multicylinder oil engines, starting valves are put on only two or three, and not more than one-half, of the cylinders; the cylinders without starting valves begin to fire first.

Figure 30-4 shows a mechanically operated air-starting valve commonly used on oil engines. No stuffing boxes are used on the stem as there is always a danger of causing the valve to stick. With a good ground fit, with labyrinth grooves on the stem, the leakage of compressed air is very small. The large diameter d_1 of the stem is made the same as the small diameter of the valve-seat cone, both being equal to about $0.13D$ in small engines and down to $0.10D$ in large engines, where D is the cylinder bore. The size of the air pipe is made either equal to or slightly larger than d_1. The maximum spring compression should be equal to the starting air pressure times the projected area of the valve seat.

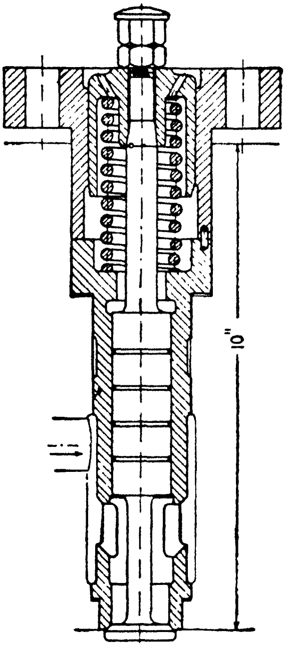


FIG. 30-4.—Air-starting valve.

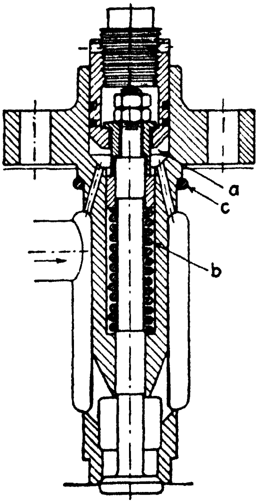


FIG. 30-5.—Automatic air-starting valve.

To simplify the air-starting mechanism in larger oil engines the valve is engaged by compressed air admitted to a small air cylinder *a*, Fig. 30-5. The tension of the spring *b* is such that, when the pressure in the cylinder exceeds the air pressure, the valve stays closed; *c* is a rubber-ring gasket. With such a valve an oil engine can be started without shutting off the fuel injection, and easier starting of the engine results. Figure 30-6 shows an indicator card obtained with this type of valve, designed for 285 psi air pressure. After the first firing, compressed air is again admitted when the expansion drops below 285 psi. The area of the card is much larger than the one obtained from compressed air or fuel injection alone.

In the case of starting valves operated by compressed air, the valve in the cylinder head becomes simply a check valve,

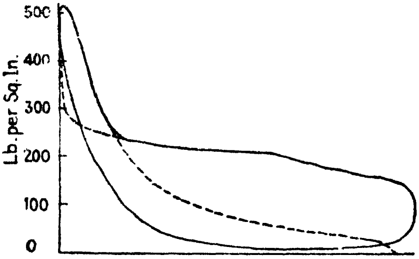


FIG. 30-6.—Air-starting diagram.

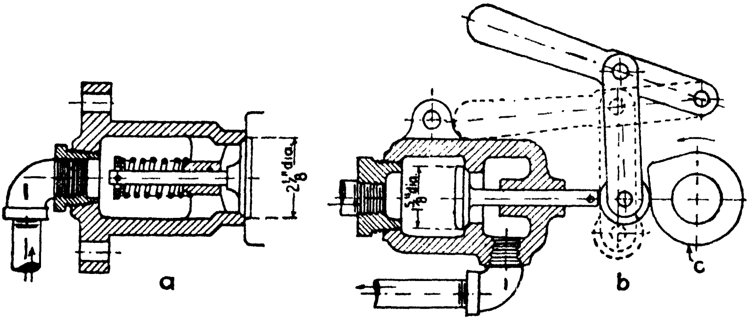


FIG. 30-7.—Air-operated starting valve.

Fig. 30-7*a*, and the air admission is controlled by a separate valve, either in the shape of a poppet valve, operated by a cam *c*, 30-7*b*, or, in the case of multicylinder engines, by a rotating distributor. A slot in the distributor disk or sleeve admits air successively to a number of ports, connected by pipes to the individual cylinders.

Oil-engine Igniters.—Small high-speed precombustion-chamber oil engines usually require some special igniter for starting when the engine is cold, because the temperature rise through compression is not sufficient to ignite the first fuel charge. Such igniters are made in the shape of an electrically heated coil or of a punk which is made to glow and then inserted through a special plug into the compression chamber, Fig. 22-25. An electric glow plug used in some high-speed compression-ignition oil engines is shown as *p* in Figs. 11-11*a* and *c*. It consists of a high-resistance wire coil heated by a 2-volt storage battery and protruding into the combustion chamber. The glowing plug heats the air in the combustion space and thus shortens the period required to raise its temperature to that necessary to ignite the injected fuel. This system is particularly helpful in cold-weather starting.

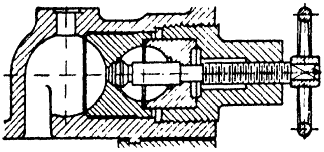


FIG. 30-8.—Dual-compression chamber of Lister engine.

In some compression-ignition oil engines cold-weather starting is facilitated by heating the air intake with a gasoline torch.

Another method consists in raising the compression ratio until the engine begins to fire. In the Lister engines, Fig. 30-8, the combustion chamber is divided into two spaces by a wall with an opening closed by a valve. When the engine is running, the valve is in its extreme right-hand position and the compression ratio is 15:1. Before starting, the valve is screwed to the left by means of the handwheel. The compression space is reduced and the compression ratio increased to 19:1, which results in a compression temperature sufficient to start the engine from cold in any weather. For starting, the exhaust valve is kept open by means of a lifter while the engine is being spun. When the lifter is released, the engine is kept going by the flywheel, and the high compression ignites the very first fuel charge. After a few minutes of operation the valve is moved to the running position. The same method is used in the National-Superior oil engines, Fig. 11-14, by turning the plug *e* 90°.

Starting Position.—From the foregoing it is obvious that only engines having five or more working cylinders, each one equipped with an air-starting valve, can be started from any position of the crankshaft. All other engines must be turned to such a position that one of the pistons is on the downward stroke, slightly past the dead center and ready to take the starting air.

In small engines the crankshaft is turned to the necessary position, by turning the flywheel either directly by hand or by means of a bar inserted into holes on the flywheel rim. Larger engines are either turned by a worm which is engaged during this operation with teeth cut in the flywheel rim, or by a ratchet mechanism, or jack, Fig. 30-9. In large engines the worm may be operated by an electrical or compressed-air motor and the ratchet mechanism by a reciprocating compressed-air motor.

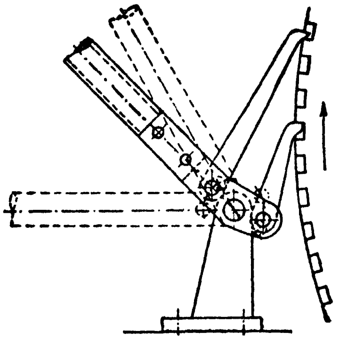


FIG. 30-9.—Flywheel barring mechanism.

30-4. Reversing.—From about 200 hp up, many marine and some locomotive oil engines are built direct reversible. A reversing gear must be simple, quick in action, and reliable. The gears actually in use fulfill these requirements and fall into two well-defined classes: those with a sliding camshaft and those with a movable link system.

Sliding Camshaft.—This type is used rather extensively for four-stroke engines, although it is equally applicable for two-stroke engines. For the operation of each valve, intake, exhaust, and starting, and of the fuel injection, there are provided two cams, one ahead and one astern, fastened to the camshaft side by side. Reversal is effected by sliding the camshaft a few inches endways in its bearings so that the ahead cam is removed from underneath the cam roller and replaced by the astern cam and vice versa.

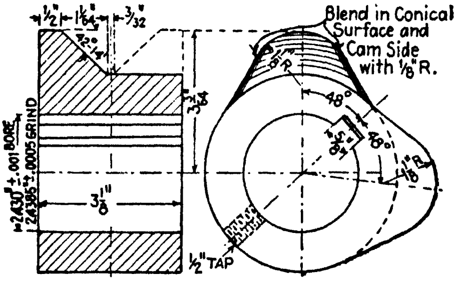


FIG. 30-10.—Intake or exhaust cam of a direct-reversible engine.

In smaller engines up to 500 or 600 hp, the cam lobes are beveled, Fig. 30-10, and the rollers have rounded edges to slide up and down between the ahead and astern cams, respectively.

In larger engines it is necessary to provide means for lifting the cam rollers clear of the cam lobes during the longitudinal movement of the camshaft.

The force required to overcome the friction when moving the camshaft endways is about one-third of the weight of the camshaft with all cams and gears keyed to it. However, it is advisable to allow a fair margin of power, as the resistance is always a matter of some uncertainty. If the rollers are not lifted by separate means, the resistance due to opening the maximum possible number of valves simultaneously must be added to the necessary axial force.

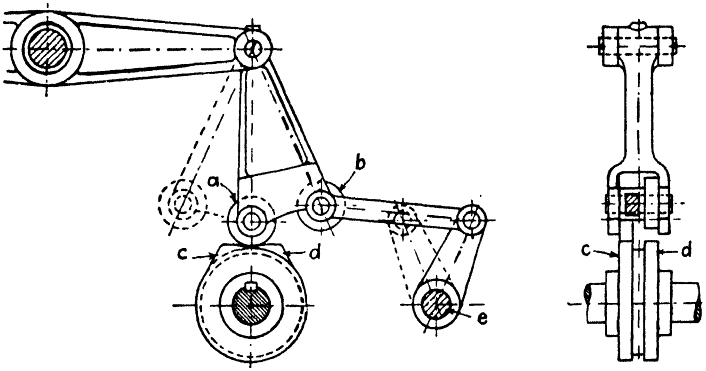


FIG. 30-11.—Roller-type reversing gear.

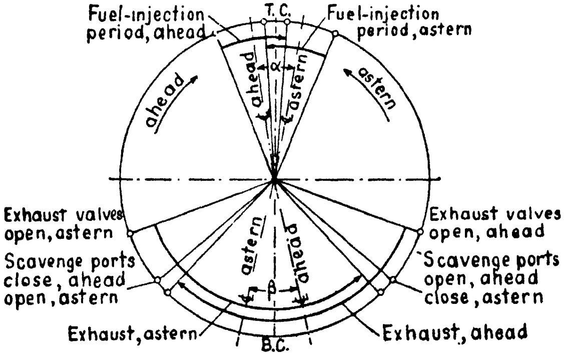

Fuel-injection period, ahead
T.C.
Fuel-injection period, astern
ahead
astern
α
ahead
astern
Exhaust valves open, astern
Scavenge ports close, ahead open, astern
Exhaust valves open, ahead
Scavenge ports open, ahead close, astern
astern
ahead
β
Exhaust, astern
B.C.
Exhaust, ahead

FIG. 30-12.—Timing diagram of a direct-reversible two-stroke oil engine.

Movable Link Gears.—Of the many schemes only the twin-roller type will be mentioned, Fig. 30-11. Rollers *a* and *b* lie in planes of the ahead cam *c* and the astern cam *d*, respectively. In

Fig. 30-11 the valve is shown controlled by the ahead cam *c* and roller *a*, roller *b* being outside the radius of action of its cam. Turning the lay shaft *e* through a certain angle throws roller *a* out of action and brings roller *b* into action with the astern cam. This type of gear is used for both four- and two-stroke engines.

Two-stroke Engines.—Most of the above described reversing gears can be and actually are used in two-stroke engines as well. However, there are available also some different schemes in which the expensive duplication of cams is avoided. Taking as an example an engine with scavenge ports and exhaust valves and neglecting temporarily the starting-air valves, the timing diagrams for ahead and astern will be along the lines of Fig. 30-12. For both fuel pump and exhaust valves all that is necessary to effect reversal is to turn the fuel-pump camshaft through an angle α and the exhaust-valve camshaft through an angle β. However, it is more usual to reverse only the exhaust valves by turning the camshaft and to use independent means, such as duplicate cams, for the fuel pumps and starting valves. Of course, the effect of turning the camshaft on the timing of the latter must be taken into consideration in fixing the angular positions of their cams.

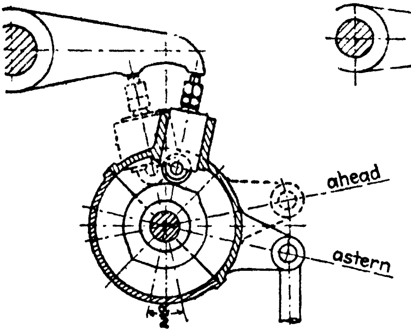


Fig. 30-13.—Reverse gear with movable roller.

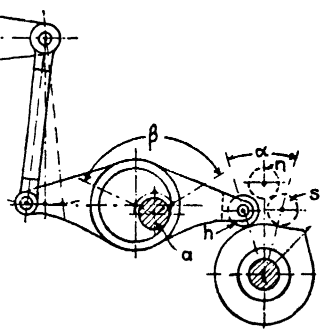


Fig. 30-14 —Reverse gear with eccentric motion.

Instead of turning the camshaft through a certain angle relative to the cam roller, the same effect may be obtained by turning the roller relative to the camshaft. One of such arrangements is shown in Fig. 30-13. The valve lift in the astern position is less than in the ahead one on account of the changed leverage, but that is not important. It is convenient to reduce the reversing angle by one-half by using a half-speed camshaft and double-nosed cams.

Another gear based on the same principle is shown in Fig. 30-14. By turning the eccentric fulcrum *a* through an angle β the roller *h* is displaced from the ahead position *h* to the astern position *s*. In the neutral position *n* the roller is outside of the radius of operation of the cam.

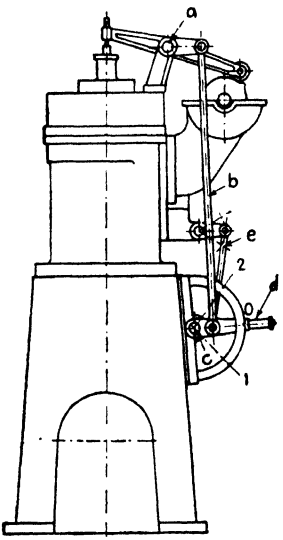


Fig. 30-15.—Oil-engine maneuvering gear.

The main trouble with reversing gears of high-speed two-stroke engines is their noisiness and wear. The sliding camshaft probably remains the simplest and best solution.

30-5. Maneuvering Gear.—The term maneuvering gear is applied to the mechanism, apart from the reversing gear, which comes into operation in starting and stopping a compression-ignition oil engine, stationary, automotive, or marine. The maneuvering gear comprises the apparatus for opening and shutting off the starting air and cutting in and out the fuel oil. These two manipulations must be properly timed and so interconnected with the reversing gear that the whole procedure of starting an engine, stopping it before reversing, and the following starting in reversal may be effected by the movement of a single lever or handwheel.

In larger engines the maneuvering gear is operated by a servomotor, usually an air motor, sometimes an electric motor.

Figure 30-15 shows a simple type of maneuvering gear for a stationary engine. The fuel lever and the starting levers are mounted eccentrically on fulcrum shafts *a* similar to Fig. 30-14. Each shaft *a* is connected by levers and links *b* to a shaft *c* with a hand-control lever *d*. In the lower position 1 of lever *d* the starting valves are in operation and in the upper position 2 all fuel valves. A link *e* connects lever *d* to an eccentric fulcrum on the fuel pump by means of which the fuel suction valves are lifted as long as the starting valves are in operation. In air-injection engines a second similar link controls the air-injection valve.

cutting it off in the neutral or stop position 0 of the maneuvering lever.

The maneuvering gear for a direct reversible engine is similar to that just described, but the hand lever must have five instead of three positions: from operating ahead to starting ahead, then neutral, or stop, position, next starting astern and the other extreme position—operating astern. The maneuvering lever also will be interconnected with the reversing gear.

Engines with electrical spark ignition do not require a maneuvering gear.

30-6. Problems.—1. A radial seven-cylinder aircraft engine, rated 175 hp at 2300 rpm, must be started by an inertia starter. Knowing that the engine speed must reach 150 rpm in order to start firing, compute the necessary starter flywheel effect Wk^2 if a flywheel speed of 12,000 rpm can be attained through a suitable train of gears.

2. A radial nine-cylinder compression-ignition aircraft engine, rated 270 hp at 2200 rpm, must be started by an inertia starter. Knowing that the engine speed must be at least 180 rpm in order to start firing, determine: (*a*) the necessary starter flywheel effect Wk^2 if a flywheel speed of 22,500 rpm can be attained through a suitable train of gears; (*b*) the hp of the electric motor, and (*c*) the number of cells and ampere capacity of the 24-volt storage battery. Use Fig. 30-2 for the motor efficiency, and estimate the speed ratio of the motor-to-inertia flywheel drive.

3. An automobile engine is rated 105 hp at 3750 rpm and requires a spinning speed of 100 rpm to start firing. The pitch diameter of the flywheel-rim gear is 19.000 in.; the Bendix pinion has a pitch diameter of 0.90 in. and a diametral pitch of 10. Assume the efficiency of the electric motor and the voltage as given in Fig. 30-2. Determine: (*a*) the necessary ampere and watt input; (*b*) the maximum hp of the motor, and (*c*) the hp which it must develop at the moment when the engine begins to fire; (*d*) present the data in the form of curves, similar to Fig. 30-2.

4. A gasoline-truck engine is rated 85 hp at 2400 rpm and requires a spinning speed of 100 rpm to begin firing. The pitch diameter of the flywheel-rim gear is 22 in. and has a diametral pitch of 8; the Bendix pinion has 11 teeth. Assume the efficiency of the starting motor and the voltage as given on Fig. 30-2. Determine: (*a*) the ampere and watt input, (*b*) the horsepower of the motor; (*c*) present the data in the form of curves similar to Fig. 30-2.

5. A compression-ignition two-stroke 4¼-in. × 5-in. four-cylinder tractor engine must be started by an auxiliary single-cylinder four-stroke gasoline engine. The normal full-load speed of the oil engine is 1450 rpm; its idling speed is 400 rpm; its starting speed is 275 rpm. The gasoline engine drives the oil engine through a gear train with a reduction 6:1 and a friction clutch. The full-load mep of the oil engine is 63 psi. Determine the necessary hp, bore, stroke, and speed of the gasoline engine.

6. Draw a timing diagram for the air-starting valve of a single-cylinder

12-in. × 16-in. horizontal four-stroke natural-gas engine. Its mep is 76 psi at full load and a normal speed of 240 rpm. The engine must turn at least at 100 rpm in order to start firing. Assume that an initial air pressure of 250 psig will be used. Draw an indicator diagram for the starting operation. Determine the necessary size of the air-starting tank.

7. Draw a timing diagram for the air-starter valves of a vertical four-cylinder four-stroke natural-gas engine. The engine develops 200 bhp at 400 rpm. It must turn at about 125 rmp in order to start firing. Assume an initial air pressure of 220 psig. Draw an indicator diagram for the starting operation, assuming that only two cylinders are equipped with starting valves. Determine the size of the starting-air tank, if the engine cannot be started with an air pressure below 125 psig.

8. Draw a timing diagram for air starting of a vertical five-cylinder four-stroke compression-ignition oil engine. The engine develops 375 hp at 450 rpm. The engine must turn at 125 rpm in order to start firing. Use an air pressure of 275 psig in the tanks. Draw an indicator diagram for the starting operation, assuming that only three cylinders have starting valves. Determine the size of the air-starting tanks, assuming a minimum starting pressure of 150 psi. Determine the necessary size of the piping between the tanks and the engine.

9. Draw an air-starting timing diagram for a vertical six-cylinder two-stroke compression-ignition oil engine. The engine develops 750 hp at 257 rpm. The engine must turn at 95 rpm in order to start firing. Use an air pressure of 300 psig in the tanks. Use the procedure shown on Fig. 30-6, and draw an air-starting diagram, assuming that only three of the cylinders have air-starting valves. Determine the dimensions and the number of the air tanks, assuming a minimum starting pressure of 175 psig.

10. Draw an air-starting diagram for a seven-cylinder four-stroke compression-ignition oil engine that develops 875 hp at 327 rpm. The engine speed must be at least 100 rpm in order to begin firing. Use an air pressure of 325 psig in the tanks; find the minimum starting pressure; determine the dimensions of the air tanks, if three tanks are to be used.

11. Draw an air-starting timing diagram for a six-cylinder four-stroke compression-ignition marine oil engine. The engine develops 720 hp at 260 rpm and its direct-reversible. The engine speed must be 80 rpm to begin firing. Use an air pressure of 300 psig in the starting tank and 1500 psig in three storage tanks for filling up the starting tank during maneuvering. Determine the necessary capacity of all four tanks if the engine must be able to be reversed 12 times with runs of a few seconds between, not enough to recharge the tanks.

12. Work problem 11, assuming that the engine is of the two-stroke type and that the storage tanks can carry a maximum pressure of 1200 psig.

CHAPTER 31

RATING AND TESTING

31-1. Power Rating. *Methods.*—There are three main methods of defining the rated power, or the horsepower by which an engine is designated.

1. By the maximum load which an engine can carry continuously.
2. By the maximum power which an engine can develop.
3. By a figure computed from a conventional (arbitrary) formula.

Maximum continuous load is determined also in different ways. At present most engine builders base this load, called *normal full load*, on the mean effective pressure, mep or p_e. In stationary gas and oil engines it is commonly considered that at the rated load, p_e should not exceed 80 psi in four-stroke engines. Data for other engines are given in Table 17-1. In two-stroke engines, it depends upon the scavenge method used and scavenge efficiency obtained. Table 15-1 gives average usable data.

Naturally, it must be remembered that, in four-stroke engines, p_e is referred to the compression and expansion strokes only and, when comparing with two-stroke engines, p_e must be divided by 2.

Only very few low-speed engines, such as large marine oil engines and blast-furnace gas engines direct-connected to blowers, are sometimes rated on the basis of the mean indicated pressure p_i and their rating given in indicated horsepower, whereas the rating of most engines is given in brake horsepower.

Every engine should be able to carry a certain overload, 10 to 20 per cent, but only in a case of emergency and not over a few hours at a time. Engines with a great excess of air, as mechanical-injection oil engines, are often capable of carrying an overload from 30 to 40 per cent. However, under such a load the combustion becomes incomplete, with smoke in the exhaust, and danger arises of injuring the engine through resulting abnormally high temperatures and heat stresses.

Another method, used by designers, of rating an engine by the piston displacement v in cubic inches per horsepower per second is basically identical with rating it by p_e. A displacement of $v = 80$ cu in./sec-hp gives practically the same rating as a mean effective pressure of $p_e = 80$ psi.

Maximum power which an engine can develop is sometimes used as its rating, even though the engine cannot develop it sustainingly. This is done in automobile engines, because, by the nature of the load carried, they do not have to develop their maximum power for any length of time.

Small stationary and tractor engines are sometimes rated on a similar basis with the difference that their rated horsepower is given equal to 90 per cent of the maximum horsepower, leaving 10 per cent for eventual overload.

Aircraft engines have several ratings: *take-off* power, *rated* power, and *cruising* power. Take-off power is the maximum power that the engine can develop for a short period, not over 5 min. Rated power is the maximum power that may be used for extended operation and is equal to 85 to 90 per cent of the take-off value. The cruising power is the power that is used for level-flight operation and is usually 70 per cent of the rated power or less. In engines with superchargers it is customary to indicate the altitude at which the power can be developed.

The rated horsepower of engines used in the oil industry, according to the API rules, is considered as 65 per cent of the maximum power, leaving an unusually large margin, about 50 per cent, for temporary overloading.

Nominal horsepower is a term used for rating automobile engines, usually for the purposes of taxation, sometimes for comparison. The nominal horsepower N of a four-stroke engine in this country is given by the NACC (National Automobile Chamber of Commerce) formula

$$N = 0.4id^2 \tag{31-1}$$

where d is the piston diameter, in.,
i is the number of cylinders.

This formula is based on a piston speed of 1000 fpm and a mean effective pressure $p_e = 67.2$ psi. Therefore the approximate actual horsepower N_a can be found by multiplying the above expression by the ratio of actual piston speed c to 1000,

$$N_a = 0.0004icd^2 \tag{31-2}$$

This would differ from the correct figure in as much as p_e differs from 67.2. The assumption involved in formula (31-1), that the horsepower output is independent of the length of the stroke, is incorrect. Long strokes, large valves, and light reciprocating parts have raised the piston speed at maximum engine output to 2500 fpm and higher and the mean effective pressure to about 80 psi.

A better approximation than by formula (31-1) is given by the formula used in Great Britain

$$N = 0.45i(d - 1.18)(l + d) \tag{31-3}$$

where l is the stroke in inches.

31-2. Performance Curves. *Variable Speed.*—For engines which operate at variable speeds, the rated horsepower at a certain speed does not give enough information. The necessary information is obtained from performance curves which give the horsepower as a function of the speeds used. In the same diagram usually are plotted some additional characteristic curves, such as brake mean effective pressure p_e, torque T, fuel consumption in pounds per horsepower-hour, and sometimes mechanical efficiency and indicated horsepower. All these data must be obtained from an actual test of the engine.

Torque.—The relation between horsepower N_b and torque T in pound-feet is

$$N_b = Tn/5252 \tag{31-4}$$

where n is the engine speed, rpm. On the other hand, the expression of horsepower for a four-stroke engine with i cylinders is

$$N_b = p_e l \pi d^2 n i/(12 \times 4 \times 2 \times 33{,}000) \tag{31-5}$$

Equating expressions (31-4) and (31-5) and solving for T give

$$T = 0.0052p_e ild^2 \tag{31-6}$$

or

$$p_e = 192T/ild^2 \tag{31-7}$$

Since for a given engine ild^2 is constant, equations (31-6) and (31-7) show that T is directly proportional to p_e. Thus for variable-speed engines both p_e and T may be represented by the same curve, only using different scales.

For two-stroke engines the corresponding expressions become

$$T = 0.0104 p_e i l d^2 \tag{31-8}$$

or

$$p_e = 96T/ild^2 \tag{31-9}$$

Performance Curves.—A typical set of performance curves for an industrial 60-hp gasoline engine is shown in Fig. 31-1. It shows that in this engine the maximum p_e or torque T is developed at 1200 rpm. However, the peak horsepower-output occurs at a considerably higher speed, 2100 rpm, after which the power begins to drop. The fuel consumption is influenced by the revolutions per minute, and its minimum value of 0.63 lb/hp-hr occurs at the rated power of the engine.

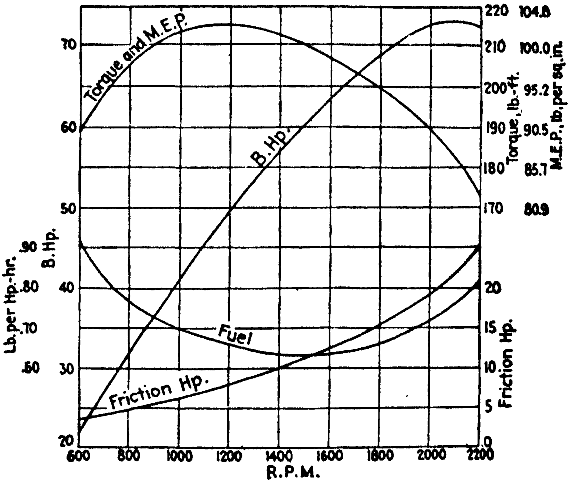


FIG. 31-1.—Performance curves of a $4\frac{1}{4}$-in. by $5\frac{1}{2}$-in. four-cylinder industrial engine.

Performance curves of automotive engines are similar to Fig. 31-1 but differ from it in that an automobile engine has to operate at different speeds and under different loads and that there is no definite relation between its load and speed. Performance curves of automobile engines may be seen in Figs. 31-9 and 31-10.

Performance curves of a radial air-cooled airplane engine are given in Fig. 31-2. The trend is the same as in Fig. 31-1 except that the range of speeds is smaller and the brake horsepower curve does not reach a peak.

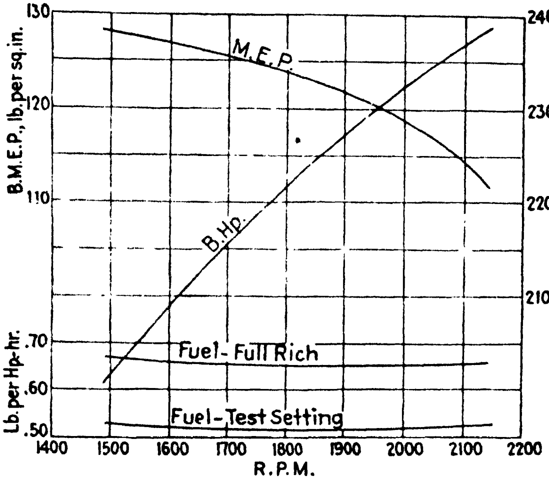


FIG. 31-2.—Performance curves for a Wright model J-4B Whirlwind engine

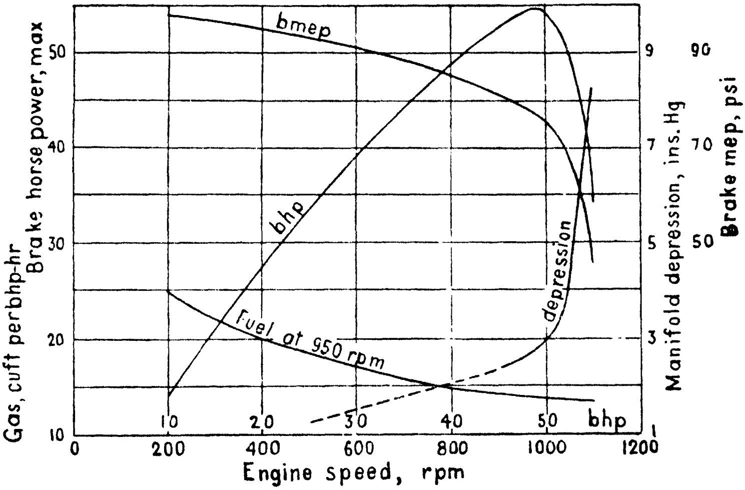


FIG. 31-3.—Performance curves of a 45-hp natural-gas engine at constant and variable speeds.

The performance curves can be improved and the peak load shifted to higher revolutions per minute and raised by supercharging, as has been discussed in Chap. 19.

Figure 31-3 shows the performance curves of a 5¼-in. × 6½-in. four-cylinder four-stroke natural-gas engine. The maximum horsepower is determined by the very abrupt increase of the manifold depression. Up to the critical speed of about 970 rpm the bmep curve runs unusually flat. The fuel consumption was determined only for constant-speed operations at 950 rpm. The fuel-consumption is referred to a gas of 1000 Btu per cu ft; the corresponding horsepower is indicated by the upper abscissa scale.

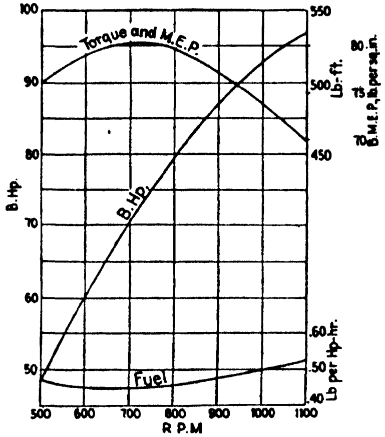


FIG. 31-4.—Performance curves of a four-cylinder oil engine.

Performance curves of a small, high-speed compression-ignition oil engine are given in Fig. 31-4. The main difference of the torque curve, as compared with torque curves of gasoline engines, lies in the fact that it is flatter and higher at low speeds; also the peak load is beyond the range of regular engine use. The higher torque at low speeds allows the use of a smaller oil engine when substituting it for a gasoline engine.

TABLE 31-1.—TEST RESULTS OF A SUPERCHARGED MARINE DIESEL ENGINE

Speed, rpm n	Brake horse-power N_e	Indicated horse-power N_i	Mech. efficiency, % η_m	Brake mep, psi	Mean indicated pressure, psi	Fuel consumption, lb/bhp-hr
115	2850	3560	80.0	85.4	106.7	
116	3100	3820	81.1	91.8	113.2	0.364
120	3380	4120	82.0	96.8	118.0	0.360
124	4190	4950	84.6	116.0	137.2	0.412
127	4450	5350	85.0	121.0	142.3	

Finally, Table 31-1 gives the performance data of a low-speed marine oil engine. The steady increase of the mep with an increase of the speed is due to supercharging. Notable also is a steady increase of the mechanical efficiency.

31-3. Testing.—A number of textbooks and manuals exist, which give information on testing internal-combustion engines; a person conducting tests should be familiar with one of these. In the following paragraphs only a few important points in conducting the tests will be outlined.

First, the object of a test must be definitely established. A test may be conducted either to find the power which the engine can develop under certain conditions, or to determine the fuel economy, or to obtain still more detailed information so as to compute a heat balance, and possibly to duplicate special conditions of actual engine operations.

Regardless of what the object of a test may be, it is always very desirable to follow the instructions of standard Power Test Codes laid down and published by the ASME. When testing an automobile engine it is advisable also to follow the Rules of the SAE, which differ only slightly from the ASME Codes, and for airplane engines to follow the Rules of the Department of Aeronautics of the U.S. Bureau of Standards.

Brake Horsepower.—Regardless of what kind of a brake or absorption dynamometer is used—a prony brake with friction between cast iron and wood, or a hydraulic brake with water friction, or an air-fan dynamometer, or finally an electric eddy-current brake with friction of magnetic lines—in all cases the same general formula is applicable:

$$N_b = 2\pi LFn/33{,}000 \qquad (31\text{-}10)$$

where L is the length of beam, ft,
F is the net brake load, lb,
n is the speed, rpm.

The factor $2\pi L/33{,}000$ is called the brake constant c, and formula (31-10) becomes

$$N_b = cFn \qquad (31\text{-}11)$$

It is convenient to make the beam length $L = 5.25$ ft, as then $c = 0.001$. For small engines, L may be made 31.5 in. giving $c = 0.0005$; and for large engines, L is made 10.5 ft. or 15.75 ft with $c = 0.002$ or 0.003, respectively.

In designing a prony brake with wooden blocks it is advisable not to exceed the peripheral speed of 2500 fpm to avoid the danger of the blocks taking fire. The face of a water-cooled drum may be determined from the condition that the drum surface must be at least 0.1 sq ft/bhp.

An electric load is very convenient when testing a constant-speed engine, provided that the efficiency curve of the generator is known and the generator is connected to the engine either directly or through a drive whose efficiency can be determined accurately.

The influence of atmospheric pressure and altitude above sea level, of temperature and humidity upon the power developed, have already been discussed in Chap. 12 and may be taken into account, using expressions (12-22) to (12-26).

Indicated Horsepower.—The general expression for indicated horsepower of an engine with any number of cylinders, single- or double-acting, is

$$N_i = \Sigma[p_i lan'/(12 \times 33{,}000)] \tag{31-12}$$

where p_i is the mean indicated pressure in each cylinder, psi,
l is the piston stroke, in.,
a is the net piston area, sq in.,
n' is the number of firings per minute.

The determination of the indicated horsepower of an internal-combustion engine is not so accurate as that of steam engines and gives more or less dependable results only if a number of precautions are taken.

The indicator must be of a special construction; its spring must not be subjected to the high gas temperatures; the inertia of the recording mechanism, for both pressure and volume, must be very small in order not to distort the diagram; the reducing-motion mechanism must be theoretically correct and not influenced by inertia forces; the connection between the engine cylinder and indicator must be very short and wide; at the same time, in small engines, its inner volume must be so small as not to change the compression ratio. For high-speed engines special indicators are developed, which are used in research work but are less suitable for general industrial use.

These indicators register pressure against crank angles, giving a so-called *open diagram.* From the open diagram a regular or closed indicator diagram can be constructed with the pressures plotted against piston displacement.

With internal-combustion engines an indicator is more valuable for checking the working cycle in different details than for finding the horsepower, except in low-speed large engines. The usefulness and even indispensability of indicator cards for setting the ignition or fuel injection, and the valve timing, for determining the volumetric efficiency, etc., have been illustrated throughout this book.

Efficiencies.—Various efficiencies which are of interest in many engineering problems were discussed in Chapter 16.

Fuel Measurements.—Consumption of gaseous fuels is measured either by standard gas meters or, in large quantities, by orifice meters and does not present any difficulty if the gas pressure p_1 and temperature t_1 are measured close to the meter. The volume V_0 at standard conditions, 14.7 psia and 60 F, is found from the characteristic equation

$$V_0 = V_1 p_1 520/p_0(t_1 + 460) \qquad (31\text{-}13)$$

Volatile liquid fuel can be measured by one of the following methods:

1. By actual weighing.
2. By volume.
3. By an orifice meter.

Gasoline consumption is best determined by weighing. The fuel container is placed on scales and is connected to the carburetor by a flexible hose made with synthetic rubber since natural rubber is attacked by gasoline.

Figure 31-5 shows a device for volumetric measurement of gasoline which consists of two glass or metal bulbs *a* and *b* connected by glass tubes. The float chamber *c* maintains a constant level 0 so long as the cock *d* is open. When determining the fuel consumption the cock *d* is closed, and the time for the level to drop from 1 to 2, or 2 to 3, or 1 to 3 is determined by means of a stop watch.

Figure 31-6 shows a convenient orifice meter. Fuel flows through a small orifice *a* under a constant head maintained by an overflow or float chamber *c*, and the difference *h* of static pressures before and after the orifice is measured. The fuel rate is found from a calibration curve for the given orifice and fuel. The head *h* varies with the rate of flow, and the accuracy of determination increases with the head. Therefore it is convenient to have three orifices of different sizes that can be combined to seven different areas and can take care of widely varying rates of flow.

Less volatile fuels, such as diesel fuel oil or kerosene, are best measured by weighing. Figure 31-7 shows a good arrangement:

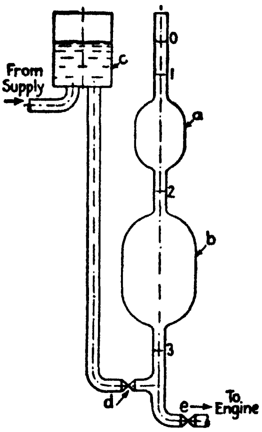


FIG. 31-5.—Gasoline measuring device.

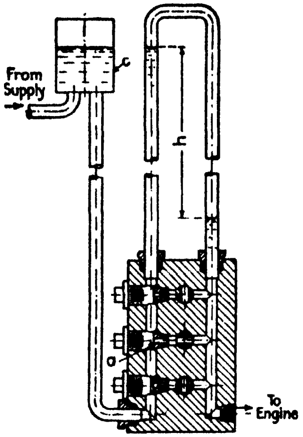


FIG. 31-6.—Yale University orifice flow meter.

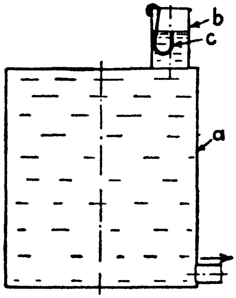


FIG. 31-7.—Fuel-oil measuring device.

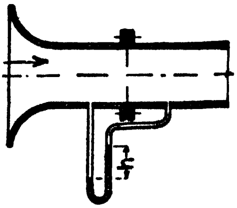


FIG. 31-8.—Air-measurement by a thin-plate orifice.

The day tank *a* has a throat *b* with a hook *c*; *a* is filled over the point of *c*, and a stop watch is started when the fuel level in coming down bares the point of *c*. The engine continues to run using the fuel from tank *a*. Toward the end of the test run, a certain weight of fuel is poured into the day tank so that its level rises again above the hook point, and, when the latter appears, the stop watch is stopped. The comparatively small cross-sectional area of *b* decreases the percentage of error in the fuel-rate determination.

Air Measurements.—A knowledge of the amount of air used may be needed in many calculations. The air may be measured by means of an orifice meter, Fig. 31-8. The weight of air in pounds per second may be determined from the equation

$$W = 18.3A\mu\sqrt{h_w w} \qquad \text{[Eq. (7-4)]}$$

where the coefficient of discharge μ depends upon the shape of the orifice and can be determined with sufficient accuracy.[1] For a sharp-edged opening in a thin plate $\mu = 0.60$. To ensure accurate results the flow at the orifice must be constant. To obtain this condition, quieting tanks must be inserted between the orifice and intake manifold of the engine.

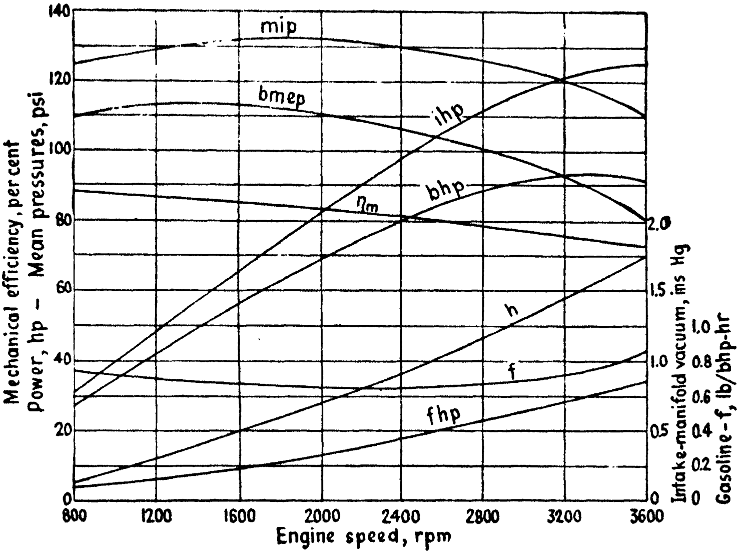


FIG. 31-9.—Performance curves of an automobile engine.

The amount of air may be determined also from the exhaust-gas analysis by finding the air-fuel ratio V_{af}, expression (4-36). This method is simple, but certain precautions must be taken to obtain accurate results.

31-4. Test Results.—Testing of an engine may be conducted to obtain many data other than horsepower and fuel consumption, data which may be valuable for the improvement of the engine or development of other engines. Such data are expediently presented in the form of curves using as the line of abscissas either the load at a constant speed or the speed for a variable-speed engine.

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[1] MARKS, L. S., "Mechanical Engineers' Handbook," 4th ed., pp. 2100–2102, McGraw-Hill Book Company, Inc., New York, 1941.

Figure 31-9 gives the curves obtained in testing an eight-cylinder $3\frac{3}{16}$-in. × $3\frac{7}{8}$-in. automobile engine with a full-throttle operation. The brake horsepower was obtained by driving an electric dynamometer; the friction horsepower was obtained by motoring the engine, using the dynamometer as a motor. From

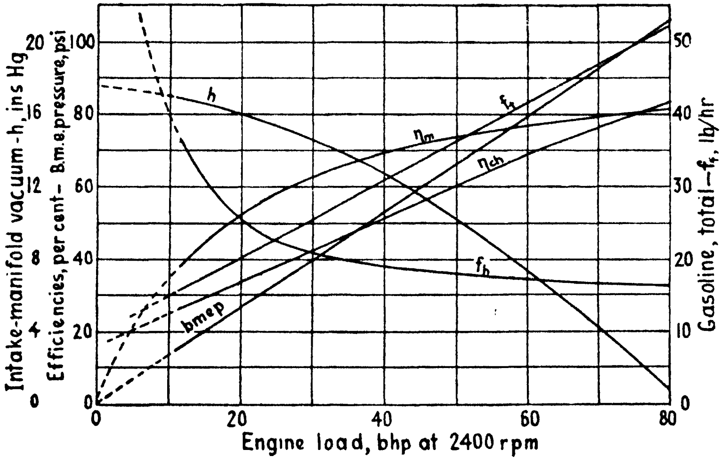

FIG. 31-10.—Performance curves of an automobile engine.

these data were found the indicated horsepower and the mechanical efficiency, as well as the bmep and the mip. The curve *f* shows the gasoline consumption, in pounds per horsepower-hour. Of interest is also the vacuum *h*, in inches of Hg, measured in the intake manifold and caused by the venturi throat. If the venturi diameter is increased to raise the maximum horsepower, the depression will become too small at the lower speeds, and the possible speed range will be reduced.

Figure 31-10 gives data obtained when testing the same engine at a constant speed of 2400 rpm and at different loads. Among other data it gives interesting information in respect to manifold vacuums produced by the throttle at different loads, its influence on the charge efficiency η_{ch}, and through it upon the mep and indirectly upon the mechanical efficiency η_m.

Heat Balance.—Energy supplied to an engine is the heat value of the fuel consumed. As has been repeatedly pointed out, only part of this energy is transformed into useful work. The rest is either wasted or may be utilized only for heating purposes.

The two main parts of the heat unavailable for work are the heat carried away by the exhaust gases and by the cooling medium. To give sufficient data for the preparation of a heat-balance sheet, a test should include the taking of indicator diagrams or some other method of determining the friction horsepower and the measurement of speed, load, fuel consumed, quantity of cooling water, its temperatures at inlet and outlet, temperature and analysis of exhaust gases. Aside from small losses, such as radiation and incomplete combustion, the above enumerated data make it possible to account for the heat supplied by the fuel and indicate its distribution.

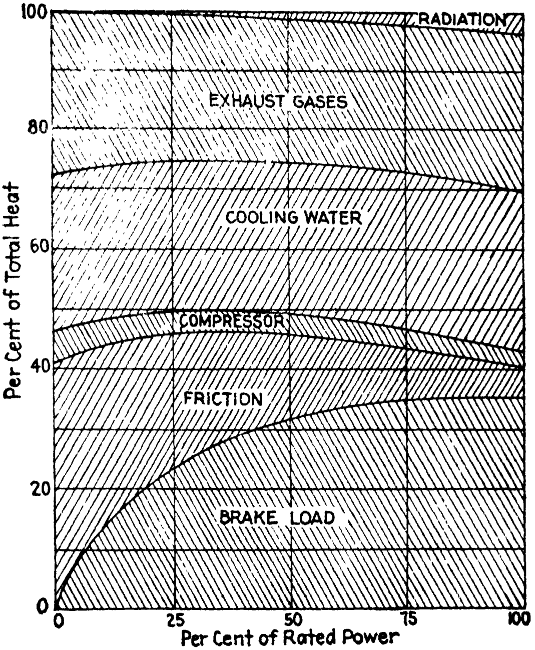

FIG. 31-11.—Heat balance of an oil engine.

Table 31-2 gives some average figures for heat balances. The large amount of heat carried away by the cooling water does not indicate poor utilization of heat inside the engine cylinder. As stated before, the relative efficiency of internal-combustion engines usually is not less than 75 and up to 95 per cent. The explanation for the large cooling-water loss lies in the fact that a large proportion of heat is transferred to the cooling water from the combustion gases after combustion is completed, especially while they pass through the cylinder head of a four-stroke engine or through the exhaust ports of a two-stroke engine. Also, practically all the piston friction work and compressor work in air-injection oil engines are included in the cooling-water heat.

TABLE 31-2.—AVERAGE HEAT BALANCES

Item	Spark-ignition engines	Compression-ignition engines
Transformed into shaft work, per cent	20–28	32–40
Carried away by cooling water, per cent	35–32	33–30
Carried away by exhaust gases, per cent	37–30	30–23
Not accounted for (radiation, etc.), per cent	8–10	5– 7
Total, per cent	100	100

If the amount of cooling-water circulation is decreased, the water will receive less heat, but the exhaust-gas temperature will rise and more heat will be carried away by the exhaust gases.

Figure 31-11 shows as an example a graphical heat balance of a four-stroke air-injection oil engine for loads from full load down to no load. The vertical distances between curves represent the corresponding percentages in heat balances for different loads. The indicated work is subdivided into brake load and friction losses; the latter increase relatively very rapidly as the load decreases. The percentages of heat carried away by the cooling water and exhaust gases vary only very little with the load.

However, they are influenced by the specific fuel consumption: The lower the specific fuel consumption, the less heat goes into the cooling medium and exhaust gases, both absolutely and in percentages of the total heat of the fuel.

Actually the heat distribution is much more involved. During the suction or scavenge period the entering fresh charge receives heat from the residual gases and from the cylinder walls. Part of the indicated work on the piston is transformed through friction into heat, which goes mostly into the cooling medium but partly into the exhaust gases and lubricating oil and also is dissipated through the crankcase to the surrounding air. A Sankey diagram of the heat balance gives a clearer understanding of the numerous heat exchanges that take place. Figure 31-12 gives such a diagram for a typical spark-ignition engine and shows how manifold and intricate the energy flow is.[1]

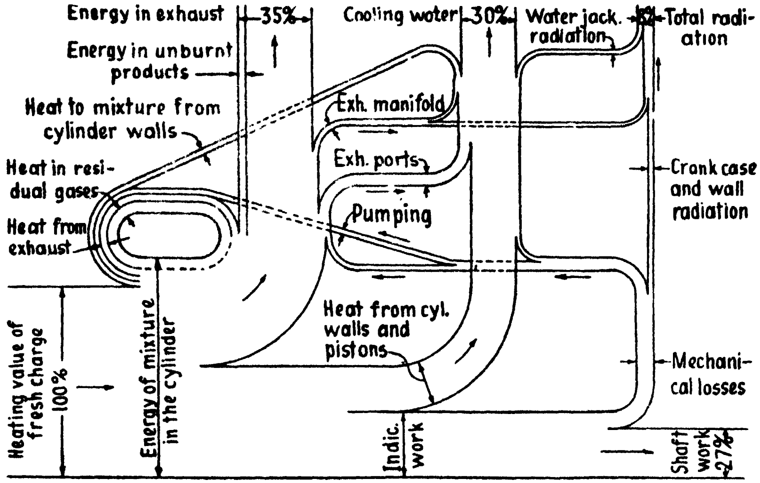

FIG. 31-12.—Sankey diagram for a typical spark-ignition engine

31-5. Problems.—1. Determine the rated horsepower of a stationary four-stroke six-cylinder compression-ignition oil engine with a 15-in. bore, 20-in. stroke, 240 rpm.

2. Determine the rated horsepower of a stationary four-stroke four-cylinder natural-gas engine with a 10-in. bore, 12-in. stroke, 450 rpm rated speed.

3. Determine the probable rated horsepower of a stationary two-stroke three-cylinder, crankcase-scavenging 12-in. × 14-in. × 360 rpm compression-ignition oil engine.

4. Determine the expected rated horsepower of a locomotive-type two-stroke eight-cylinder uniflow scavenging 8-in. × 10-in. × 660 rpm compression-ignition oil engine.

5. Determine the nominal NACC horsepower rating of an eight-cylinder automobile engine with a $2\frac{3}{4}$-in. bore, $4\frac{3}{4}$-in. stroke; compare it with an approximate maximum rating at 3000 rpm and also with the formula used in Great Britain.

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[1] LICHTY, L. C., "Internal Combustion Engines," 5th ed., p. 477, McGraw-Hill Book Company, Inc., New York, 1939.

6. Find the torque developed at full load by a 50-hp three-cylinder oil engine, $6\frac{1}{2}$-in. bore, $10\frac{1}{2}$-in. stroke, 500 rpm, and the corresponding mean effective pressure.

7. Find the load in horsepower on an engine when running at 375 rpm under a prony brake with a beam length of 48 in., unbalanced weight of the beam 15 lb, and scales showing a gross load of 180 lb.

8. Determine the mechanical efficiency of the engine in problem 7 if the engine is of the four-stroke type, has two cylinders, $8\frac{1}{2}$-in. bore, 12-in. stroke, and the mean indicated pressure is 92 psi in one cylinder and 93.5 psi in the other.

9. Find the mean indicated pressure of a six-cylinder two-stroke double-acting vertical oil engine; bore 24 in., stroke 55 in., diameter of piston rod 8 in. The power developed with high scavenge pressure is 6400 ihp at 100 rpm. The mean indicated pressures are the same in all working spaces.

10. Determine the brake thermal efficiency of a 60-hp gas engine which uses 310 cu ft natural gas in 30 min; heat value of the gas is 1015 Btu/cu ft.

11. An engine with a compression ratio $r = 5.5$ uses 0.42 lb/ihp-hr gasoline which has a heat value of 18,000 Btu/lb. Assuming a theoretical air-fuel ratio, find the indicated thermal and relative efficiencies of the engine.

12. The volumetric efficiency of a carburetor engine, based upon air measurement alone, is 87 per cent. Find the volumetric efficiency taking into consideration the volume of gasoline with a theoretical air-fuel ratio.

13. Compute the mep and torque of the engine in problems 7 and 8.

14. Compute the approximate shaft torque developed by the engine of problem 9; also the torque developed at 75 rpm if the mean effective pressure changes as the cube of the engine speed.

CHAPTER 32

GAS TURBINES

32-1. General Features.—The problem of converting the energy released by combustion into mechanical work by means of a gas turbine has occupied inventors and engine builders for many years. However, only during the last few years have gas turbines become practical, at least in some applications. Already several large engine-building concerns in this country are engaged in the development of this type of internal-combustion engine.

The difficulties that must be overcome are due partly to the corrosion of metal surfaces, chiefly of the moving turbine blades, by the high-temperature gases that contain a large amount of oxygen, and to the creep of high stressed metal parts under elevated temperatures. These difficulties are being gradually reduced by the progress in steel metallurgy. Other drawbacks are a relatively low over-all efficiency and the fact that a gas turbine is not self-starting.

Against these drawbacks a gas turbine has the important advantages of simplicity and light weight; another potential advantage is the use of pulverized coal instead of liquid or gaseous fuels. Simplicity is the same as in a steam turbine, and light weight is due to the high speeds used and to a moderate weight of the accessories and auxiliaries. The possibility of using pulverized coal depends upon cleaning it of ash-forming admixtures or upon using a special closed cycle.

The main drawback—low over-all efficiency—may be overcome by a further increase of the temperature of the working gases at the turbine, by improving the basic heat cycle, and by raising the hydraulic efficiency both of the air compressor and of the turbine proper.

Types.—Depending upon the method of combustion, gas turbines may be subdivided into two groups: (*a*) explosion turbines and (*b*) continuous-combustion turbines.

Another classification, based upon the action of the expanding gases, similar to steam turbines, is (*a*) impulse turbines, (*b*) reaction turbines, and (*c*) combination turbines with both impulse and reaction bladings.

32-2. Explosion Turbine.—In an explosion turbine, fuel is injected at *f* into compressed air admitted to a closed chamber *b*, Fig. 32-1, and the air-fuel mixture is ignited by a spark plug *s*. The pressure rise opens a check valve *c* and the gases flow expanding to atmospheric pressure, through the nozzle *n*, acquire a high velocity, and strike the blades of the turbine wheel *w*. Following the explosion a certain amount of scavenging air is admitted to the chamber through a mechanically operated valve *a* and passed through the turbine blading to cool it. After that, valve *c* closes, then valve *a*, when the pressure in the chamber *b* and spring tension equal the compressed-air pressure; and the cycle is repeated.

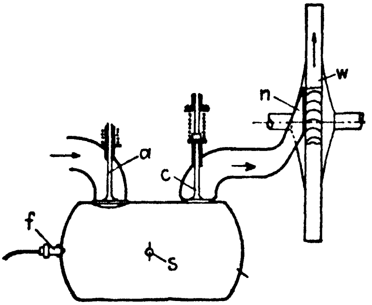

FIG. 32-1.—Explosion gas turbine.

The drawback of this cycle is an intermittent action with variable pressure differences and gas velocities. This results in low thermodynamic and hydraulic efficiencies of the turbine. Several explosion turbines were designed and built, but the results obtained were not encouraging. However, at least one large turbine of this type was reported to be in operation in the Thyssen Steel Works in Germany.

32-3. Combustion Turbine.—A basic continuous-combustion, or, briefly, combustion gas turbine, consists of three main parts: an air compressor, a combustion chamber with a fuel burner, and a gas turbine. These three elements with the necessary interconnections are shown in true relative size in Fig. 32-2. The turbine resembles a noncondensing reaction steam turbine, but has fewer stages. Mechanical power is developed by expansion of the hot, high-pressure gases in the consecutive stages of stationary and rotating blades.

Operating Cycle.—The theoretical cycle of this turbine is represented in Fig. 32-3; atmospheric air with an initial pressure p_1 and temperature T_1 is compressed isentropically to p_2 and T_2, curve 1-2. Heat is added at constant pressure p_2, and the temperature rises to T_3. After an isentropic expansion to the initial pressure $p_1 = p_4$ and a temperature T_4, the gases are cooled and replaced by fresh air. The total work done by the gas in passing through the turbine is represented in the p-V diagram by the area 5-6-3-4-5, and the work absorbed by the compressor is the area 5-1-2-6-5. The difference, area 1-2-3-4-1, is the net work of the turbine. The temperature of the gases after combustion with theoretical air would be approximately 3300 to 3600 F. Since the temperature which can be used on the turbine blading is only 1000 to 1200 F, T_3 in Fig. 32-3*b*, the gases must be cooled by admitting additional compressed air. Gases of 1200 F contain about 600 per cent excess air.

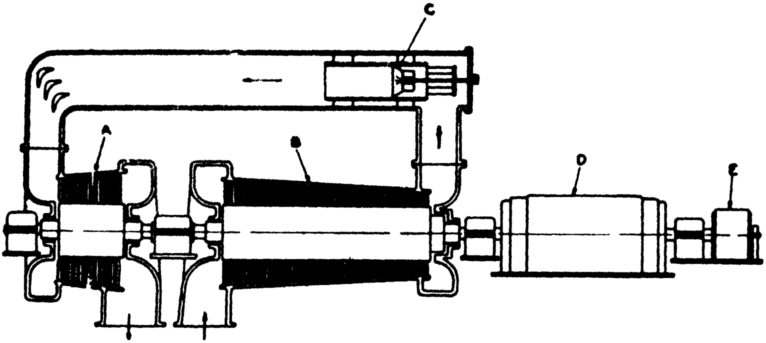

Fig. 32-2.—Combustion turbine. *A*, turbine; *B*, compressor; *C*, fuel burner; *D*, electric generator; *E*, starting motor. (*Courtesy of Allis-Chalmers Manufacturing Company.*)

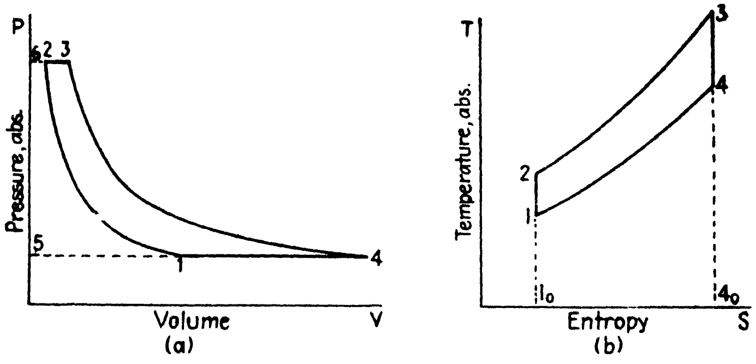

Fig. 32-3.—Theoretical cycle of a combustion turbine.

Theoretical Efficiency.—Assuming constant specific heats and neglecting the chemical changes during combustion, the ideal thermal efficiency may be presented for 1 lb of air as

$$\eta = \frac{c_p(T_3 - T_2) - c_p(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2} \tag{32-1}$$

For compression and expansion taking place between the same pressure limits the following relation exists:

$$T_2/T_1 = T_3/T_4 = (p_2/p_1)^{\frac{k-1}{k}} \tag{32-2}$$

Presenting this relation as

$$T_4/T_1 = T_3/T_2$$

noticing that

$$T_4/T_1 - 1 = T_3/T_2 - 1 \quad \text{or} \quad (T_4 - T_1)/T_1 = (T_3 - T_2)/T_2$$

and transposing the inner terms gives

$$(T_4 - T_1)/(T_3 - T_2) = T_2/T_1 \tag{32-3}$$

Therefore, from equation (32-1)

$$\eta = 1 - T_1/T_2 \tag{32-4}$$

or in connection with expression (32-2)

$$\eta = 1 - (p_1/p_2)^{\frac{k-1}{k}} \tag{32-5}$$

By substituting volumes instead of pressures from equation (2-48), equation (32-5) may be changed to

$$\eta = 1 - (V_1/V_2)^{1-k} \tag{32-6}$$

which shows that the ideal efficiency of a constant-pressure combustion cycle with a complete expansion is identical with the efficiency of a constant-volume combustion cycle with an incomplete expansion, or Otto cycle, as used in spark-ignition engines. As for the Otto cycle, the theoretical efficiency depends only on the compression ratio and is independent of the load.

Actual Efficiency.—The efficiency of the actual cycle is reduced by losses in the compressor and turbine. Part of the losses is due to friction in the bearings, but the greater part is due to loss of energy of the gases during their travel through the compressor and turbine bladings. The following designations may be used:

Q is constant-pressure heat of combustion, Btu per lb fuel;
W_c is ideal input to the compressor, Btu per lb fuel;
W_{ca} is actual input to the compressor, Btu per lb fuel;
$\eta_c = W_c/W_{ca}$ is compressor shaft efficiency;
W_t is ideal output of the turbine, Btu per lb fuel;
W_{ta} is acutal output of the turbine, Btu per lb fuel;
$\eta_t = W_{ta}/W_t$ is turbine shaft efficiency.

With these designations the over-all or actual efficiency of the installation may be presented as the ratio of the net output over the total heat input, or

$$\eta_a = (W_{ta} - W_{ca})/Q \tag{32-7}$$

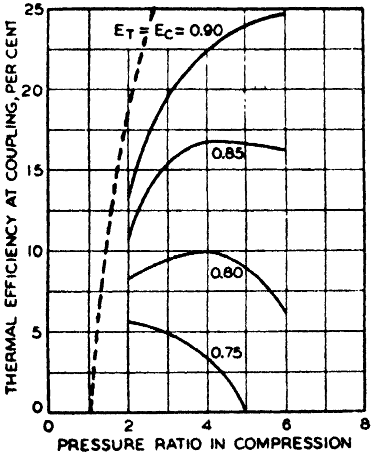

FIG. 32-4.—Effect of pressure ratio on thermal efficiency.

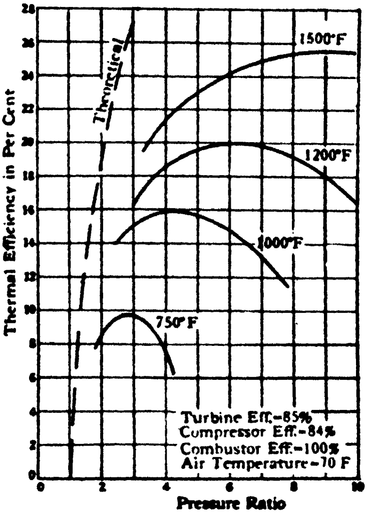

FIG. 32-5.—Effect of pressure ratio and temperature at the turbine on thermal efficiency.

Substituting for the actual output and input the ideal values and individual efficiencies and simplifying the expression gives

$$\eta_a = (W_t\eta_t\eta_c - W_c)/Q\eta_c \tag{32-8}$$

This shows that the over-all efficiency may be increased by improving either η_t or η_c or both.

Figure 32-4 shows the influence of the pressure ratio and of the shaft efficiencies of the compressor and the turbine upon the over-all efficiency.[1] For simplicity's sake it was assumed that $\eta_t = \eta_c$ for all four cases.

Figure 32-5 shows the influence of the pressure ratio and inlet temperature upon the over-all efficiency η_a computed for $\eta_t = 0.85$ and $\eta_c = 0.84$, as actually attained in some installations.

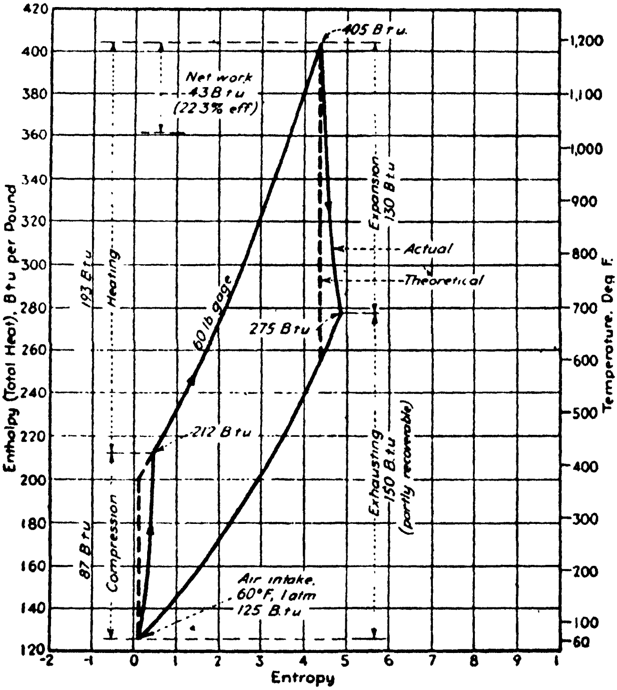

FIG. 32-6.—Enthalpy-entropy diagram of a combustion turbine.

Figure 32-6 gives a combined temperature-entropy and enthalpy-entropy diagram showing theoretical and actual cycles for a pressure ratio $p_2/p_1 = 5$ and a temperature T_3 of 1200 F at the turbine.[2]

Example 32-1.—Using data from Fig. 32-6, determine the theoretical efficiency of the installation.

By expression (32-1), with $T_1 = 60 + 460 = 520$ R,

$$T_2 = 365 + 460 = 825 \text{ R}$$

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[1] RETTALIATA, J. T., The Combustion Gas Turbine, *ASME Trans.*, vol. 63, No. 2, February, 1941.

[2] *Mech. Eng.*, vol. 66, p. 363, June, 1944.

$T_3 = 1200 + 460 = 1660$ R, and $T_4 = 590 + 460 = 1050$ R,

$$\eta = 1 - \frac{1050 - 520}{1660 - 825} = 1 - 0.635 = 0.365 \text{ or } 36.5 \text{ per cent}$$

By expression (32-8), with $\eta_t = \eta_c = 1$ and taking from Fig. 32-6

$$W_t = 405 - 255 = 150 \text{ Btu/lb} \qquad W_c = 200 - 125 = 75 \text{ Btu/lb}$$

and $Q = 405 - 200 = 205$,

$$\eta = (150 - 75)/205 = 0.366 \text{ or } 36.6 \text{ per cent}$$

a very good check.

Example 32-2.—Using data from Fig. 32-6, find the influence of the deviation of the compression and expansion lines from an isentropic, due to gas friction when passing through the bladings.

In this case $W_t = 405 - 275 = 130$ Btu/lb,

$$W_c = 212 - 125 = 87 \text{ Btu/lb}$$

and $Q = 405 - 212 = 193$ Btu/lb; therefore

$$\eta = (130 - 87)/193 = 0.223, \text{ or } 22.3 \text{ per cent}$$

The efficiency drop from 0.366 to 0.223 is almost 39 per cent.

32-4. Methods to Improve Efficiency.—While the two means of improving the efficiency, by increasing the compression ratio and by raising the inlet temperature, are very effective, as shown by Fig. 32-5, their application is limited by the decreasing resistance of the bladings to corrosion and creep.

The third way is by increasing the efficiencies of the compressor and turbine, as illustrated by Fig. 32-4. This is obtained chiefly by applying aerodynamic principles in improving the shape of the blading. By making the compressor blades airfoil-shaped and by properly spacing them, the efficiency of the axial-flow rotary compressor was brought up to 84 to 85 per cent. A further increase is difficult, but possible. By a careful design of the blading of the reaction turbine an efficiency of 88.4 per cent has been attained. Further improvement is possible.

At present, out of the total power developed by the turbine, a large part is absorbed by the compressor: about 78 per cent with an inlet temperature of 900 F, 73 per cent with 1000 F, and 67 per cent with 1200 F. This results in low over-all efficiencies of the set, although the turbine as such has a rather high efficiency.

Thermodynamic Improvements.—The three methods for improving the over-all efficiency of a gas-turbine set are: (*a*) regeneration, (*b*) reheating of the working gases, and (*c*) intercooling of the compressed air.

Regeneration.—A very effective arrangement is to preheat the compressed air, before it enters the combustion chamber, passing it over a heat exchanger and using the hot exhaust gases after the turbine as a source of heat. Preheating reduces fuel consumption and thus improves the cycle efficiency.

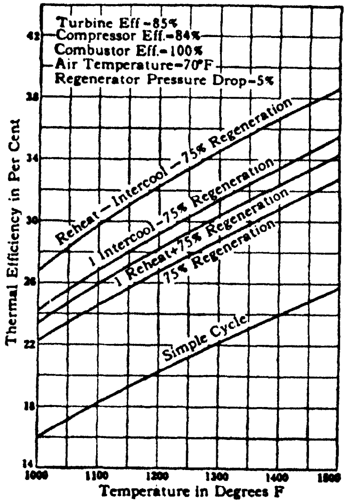

Fig. 32-7.—Influence of improving devices and temperature on thermal efficiency of a combustion turbine.

A drawback of such a heat exchanger, acting as a heat regenerator, is the low over-all heat-transfer coefficient when both the heating and heated fluids are gases. This, in turn, requires large heat-exchange areas. Theoretically, regeneration could supply approximately 22 per cent of heat required to raise the temperature of air from 313 to 1200 F. In practice, in order not to use excessively large heat exchangers, only about one-half of this amount can be reclaimed and at best not over 75 per cent. Figure 32-7 shows the gain in efficiency obtainable through regeneration. Another drawback is the resistance to flow of the gases, which causes a pressure drop in the compressed air and increases the back pressure on the turbine. However, by careful design these losses can be kept rather low.

Reheat.—By reheating or adding heat to the gases after they have passed through part of the rows of the turbine blading, a further increase in efficiency is obtained. In reheating, the gas temperature, which has dropped due to expansion, is brought back to approximately the initial temperature. Since the working gases contain about 85 per cent air, additional fuel can be burned by injecting it into the gases without any additional air supply. The gain due to reheating depends upon several conditions, as can be seen from Fig. 32-7.

Intercooling.—The work of compressing air depends upon the exponent of the compression line. If the isentropic compression is considered as a standard, then the work of compressing air in a turbo-compressor is about 20 to 25 per cent higher than this standard, as indicated in Fig. 32-6. There is a considerable saving in work of the second stage of compression if, as is done in two-stage reciprocating compressors, after compressing the air to about one-half of the required compression ratio, the air is cooled to approximately the initial temperature. Under average conditions an intercooler reduces the total power required by about 15 per cent. Figure 32-7 shows the resulting increase in the over-all efficiency.

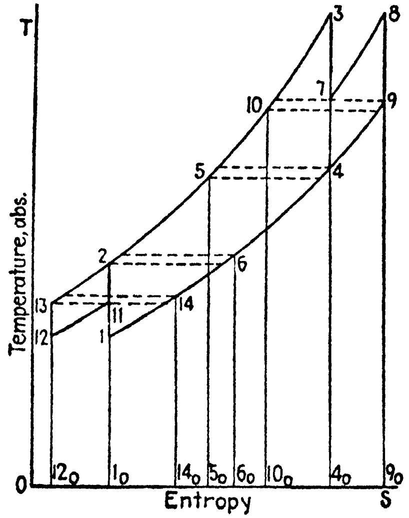

FIG. 32-8.—Temperature-entropy diagram.

Initial Temperature.—A decrease in the initial temperature of the air entering the compressor also increases the efficiency, some 0.6 to 0.25 per cent for each 10 F reduction of the temperature, depending upon the cycle used.

Entropy Diagrams.—The gains due to various additions to the simple combustion cycle can be best illustrated in the *T*-*S* coordinates, Fig. 32-8, which gives the same five combinations as shown in Fig. 32-7, only for theoretical cycles.

Simple Cycle.—The available work, or net turbine output, is presented by the area 1-2-3-4-1, which is equal to the difference

TABLE 32-1.—SUMMARY OF CONDITIONS ILLUSTRATED BY FIG. 32-8

Cycle used	Work done, net output	Outside heat added	Heat from regenerator	Heat rejected
Simple cycle	1-2-3-4-1	1_0-2-3-4_0-1_0	. . . .	1_0-1-4-4_0-1_0
Regeneration added	1-2-3-4-1	5_0-5-3-4_0-5_0	6_0-6-4-4_0-6_0	1_0-1-6-6_0-1_0
Reheat and regeneration	1-2-3-7-8-9-1	10_0-10-3-7-8-9_0-10_0	6_0-6-9-9_0-6_0	1_0-1-6-6_0-1_0
Intercooling and regeneration	1-11-12-13-3-4-1	5_0-5-3-4_0-5_0	14_0-14-4-4_0-14_0	12_0-12-11-1-14-14_0-12_0
Intercooling, reheat, and regeneration	1-11-12-13-3-7-8-9-1	10_0-10-3-7-8-9_0-10_0	14_0-14-9-9_0-14_0	12_0-12-11-1-14-14_0-12_0

between the heat added, area 1_0-2-3-4_0-1_0, and heat rejected, area 1_0-1-4-4_0-1_0.

Regeneration.—By using a heat exchanger between the hot exhaust gases and the compressed air, the temperature of the air can be raised from T_2 to T_5, where T_5 must be slightly lower than T_4 to obtain a temperature differential. The temperature of the exhaust gases will drop from T_4 to T_6, where $T_6 > T_2$ for the same reason. Naturally, the areas of heat gained by the air, 1_0-2-5-5_0-1_0, and of heat lost by the exhaust gases, 6_0-6-4-4_0-6_0, are equal. While the net output of the cycle is the same as with the simple cycle, the outside heat added is reduced, and the cycle efficiency thus is appreciably increased.

Reheat.—The net output is increased by the area 4-7-8-9-4. The heat added from the outside is represented by the area 10_0-10-3-7-8-9_0-10_0. However, the heat equal to the area 4_0-4-9-9_0-4_0 is transferred to the compressed air and regenerated, thus reducing the heat added from outside. The result is a further slight increase of the efficiency, as shown in Fig. 32-7.

Intercooling.—By cooling the partially compressed air from the temperature T_{11} down to $T_{12} = T_1$, the negative work of the compressor is reduced by the area 11-12-13-2-11, Fig. 32-8, which is therefore added to the net output.

If intercooling were used with a

simple cycle, it would increase the net output, but would require an additional amount of outside heat, area 12_0-13-2-1_0-12_0, resulting in a lower over-all efficiency. Therefore intercooling is used only in connection with regeneration. In this case the additional heat is taken from the exhaust gases, area 14_0-6-6_0-14_0, and the increase of the net output raises the cycle efficiency of the set, as shown in Fig. 32-7.

When intercooling is used in combination with regeneration and reheating, a further increase of net output and of the cycle efficiency is obtained, as shown in Fig. 32-7.

Table 32-1 shows a summary of the improvements of the simple cycle as illustrated by Fig. 32-8.

Table 32-2 gives some additional information about the working conditions in gas turbines with various refinements of the cycle. The data are based on a constant net output.[1]

Impulse Stages.—The proposal to use, in a combustion turbine, one or two impulse stages before the usual reaction stages, has for its main object an increase of the temperature of the gases at the turbine. Since one impulse stage corresponds to a temperature drop of about 150 F in the nozzle, the temperature of the gases may be higher by this amount at an impulse-stage turbine than at a pure-reaction turbine, assuming for both cases the same temperature at the first row of blades, in the runner of an impulse turbine and in the stationary cylinder of a reaction turbine. Figure 32-7 shows that a temperature increase of 150 F raises the over-all efficiency from 3 to 4 per cent.

32-5. Closed Cycle.—The maximum net output rating of a practical open-cycle, single-flow combustion gas turbine, as discussed in Secs. 32-3 and 32-4, is about 10,000 hp, regardless of its speed. This limit is set by the maximum practical blade dimensions and size of piping required.

If, however, the absolute pressures of the working gases are multiplied by 10, their volumes are divided by 10 and all linear dimensions by about 2.2. In a closed cycle the working gas is recirculated and kept at relatively high pressures in order to reduce the dimensions of the compressor, turbine, and piping.

A basic closed-cycle installation consists of a compressor, a gas heater, a turbine, and a gas cooler. In the compressor the initial pressure of the working gas, which may be air, hydrogen, helium, or some other gas, is increased from about 150 psia in a ratio of about 4:1. The high-pressure gas is heated by an external gas heater, expanded in the turbine, and cooled back to the initial temperature and pressure in a cooler. The gas heater is a heat exchanger between the high-temperature combustion gases and the high-pressure working gas. The exhaust combustion gases may be used to preheat the combustion air. The gas turbine is similar to one of the open-cycle type and may have

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[1] *Westinghouse Eng.*, May, 1944, p. 83.

TABLE 32-2.—EFFECT OF DIFFERENT OPEN-CYCLE ARRANGEMENTS ON EACH MAJOR ELEMENT

Item	Simple cycle	Cycle with regeneration	Intercool and regenerate	Reheat and regenerate	Intercool, reheat, regenerate
Figure No.	3	4	5		6
Input in fuel	4.95	3.75	3.43	3.55	3.11
Turbine rating	3.95	2.95	2.80	2.88	2.55
Compressor power	2.95	1.95	1.80	1.88	1.55
Useful output	1.00	1.00	1.00	1.00	1.00
Efficiency at 1200 F, per cent	20.2	26.6	29.2	28.1	32.2
Gas temperature in °F at					
Turbine inlet	1200	1200	1200	1200	1200
Leaving reheater	.....	.....	.....	1200	1200
Turbine exhaust	635	790	695	920	865
Leaving regenerator	.....	455	350	560	520
Air temperature in °F at					
Compressor inlet	70	70	70	70	70
Leaving intercooler	.....	.....	70	....	70
Leaving compressor	490	340	230	440	405
Entering combustor	490	680	575	800	750
Pressure, psia					
Compressor inlet	14.7	14.7	14.7	14.7	14.7
Compressor discharge	88.2	51.5	73.5	73.5	102.9
Turbine inlet	88.2	50.2	71.7	71.7	100.4
Turbine exhaust	14.7	15.1	15.1	15.1	15.1

Tabulation based on

Turbine efficiency	85%	Ambient air temperature	70 F
Compressor efficiency	84%	Regenerator pressure drop	5%
Combustor efficiency	100%		

any or all refinements of that cycle. The gas cooler is a heat exchanger with water circulation. The appearance of the *T-S* diagram does not differ from that of the open cycle. However, all heat rejected must be given up to the water going through the cooler.

Hydrogen in a closed cycle presents the advantage of a higher heat conductivity, about 6.8 times that of air, and therefore requires considerably smaller areas of all heat exchangers—gas heater, regenerator, and cooler.

In general the closed-cycle turbine installation is similar to a steam turbine plant: The gas heater takes the place of the steam boiler, the compressor that of the feed-water pump, and the gas cooler that of the condenser. The practical advantage is in the absence of many steam-plant auxiliaries and the theoretical advantage in the absence of the latent heat of evaporation of the working substance.

A decided advantage of the closed cycle, as compared with the open one, is in that the products of combustion do not come in contact with the blades of the turbine. This simplifies materially the problem of using coal as a fuel. Economically a cycle using a cheaper fuel and having an indifferent thermal efficiency may be superior to a cycle having a considerably higher thermal efficiency but using a more expensive fuel.

Escher Wyss in Switzerland is working on the development of a closed-cycle gas turbine. In this country the Westinghouse Electric & Manufacturing Company is working on the development of a cycle with recirculation of the working gases, but with internal combustion. The main feature of such a cycle is a smaller size for the same power or a greater power for the same physical dimensions.

32-6. Gas Turbines in Operation.—Many gas turbines are already in actual operation, both in Europe and in this country;

The greatest number are exhaust-gas turbines driving centrifugal blowers for supercharging various four-stroke internal-combustion engines, as discussed in Secs. 19-2 and 19-3.

Next come gas turbines operating in connection with the Houdry catalytic cracking process of manufacturing gasoline. The arrangement is shown in Fig. 32-10*d*. The first of 36 such turbines installed in this country has been in satisfactory operation at the Marcus Hook, Pennsylvania, refinery of the Sun Oil

Company since 1937. The turbine output is 7100 hp, of which 5900 hp is required to run the compressor; the balance is used to drive an electric generator. The compressed air is needed for the cracking process, and the power developed is a by-product. Therefore economically the over-all efficiency of the installation is high.

A still larger number of gas turbines, about 70 in land power plants all over the world and many more on steam-turbine-driven ships are installed in connection with Velox boilers, as shown in Fig. 32-10*e*. In this case again compressed air is required for producing high-pressure steam, and the excess power of the turbine is a by-product. Since the gas temperature is lowered considerably when heat is given up to the boiler, the combustion may be conducted with a relatively small air excess. This reduces the size of the air compressor and results in a larger power output and a high over-all efficiency.

Another similar application of gas turbines is the production of compressed air for blast furnaces. Since the pressure of blast air is lower than required for the gas turbine, the air is taken from an intermediate stage of the compressor, as shown in Fig. 32-10*c*. Electric power again may be furnished as a by-product. At least one such installation was built by Brown Boveri in Switzerland and is in operation in a European steel mill. All the available power of the unit is taken as blast air, about 35,000 cfm at 17 psig. The exhaust gases are used to preheat the blast air by means of a large heat exchanger.

So far the only plant where a gas turbine is installed exclusively for furnishing power is the 4000-kw gas turbine built by Brown Boveri for an underground stand-by plant at Neuchatel, Switzerland.

An oil-fired gas-turbine electric locomotive was built in 1941 by Brown Boveri for the Swiss Federal Railways, Fig. 32-9. The 2200-hp net-output turbine met all requirements during the tests; however, shortage of fuel oil did not allow the use expected. Starting the locomotive takes 8 min from cold and includes bringing the gas-turbine unit up to about one-third speed, by a small auxiliary diesel unit, and warming up the gas turbine while the locomotive can move to its train at 6 mph under the power of the diesel set. Control of the locomotive during operation is very simple and almost automatic.

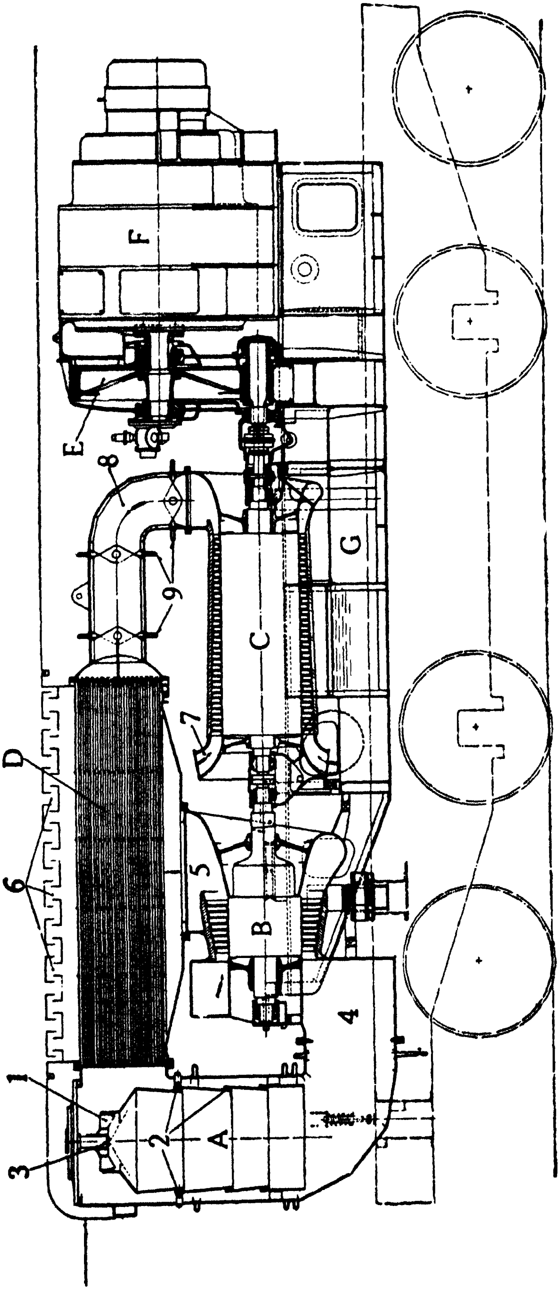

Fig. 32-9.—Locomotive with a combustion turbine as prime mover

32-7. Further Development.—While the over-all efficiency of the present gas turbines is considerably lower than that of reciprocating internal-combustion engines and probably never will equal the latter, they have great possibilities because of the great versatility which gas turbines present in respect to heat-balance arrangement, as shown in Fig. 32-10.

Arrangement (*a*) shows a basic power unit that has no auxiliary apparatus and requires no cooling water.

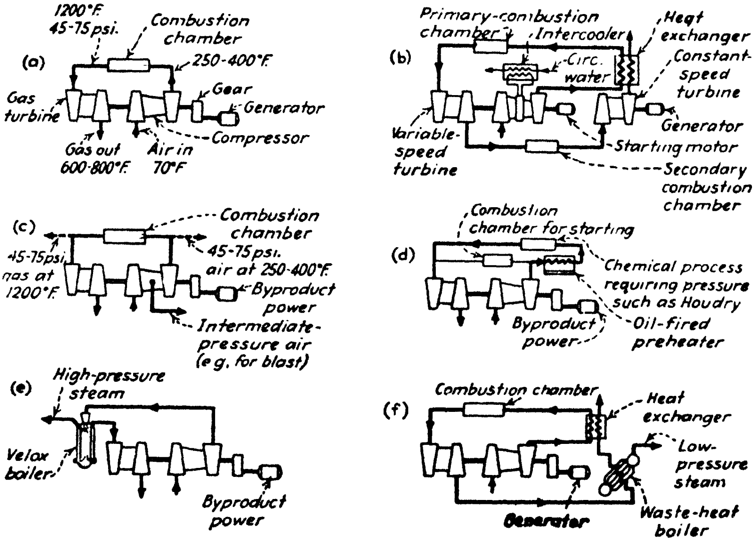

FIG. 32-10.—Various heat-balance arrangements with gas turbines.

Scheme (*b*) shows a unit with all refinements: intercooler, heat exchanger acting as a regenerator, reheater, and separate high- and low-pressure turbines. The use of two separate turbines, one to drive the air compressor and the other the electric generator, permits the regulation of the amount of air delivered by the compressor in accordance with the generator load by changing the compressor speed. This improves the partial-load efficiency of the combined unit. The over-all efficiency of this gas-turbine set, because of all the auxiliaries used, approaches that of the best steam-turbine plants. However, its initial cost is lower and its operation simpler.

Schemes (*c*), (*d*), and (*e*) have already been discussed in Sec 32-6

Scheme (*f*) shows an arrangement where the by-product steam, from waste-heat boilers, may be had in limited quantities.

32-8. Applications.—Summing up the information available about installations already in operation, those designed and being built, and future probable developments, the applications of gas turbines, outside of exhaust-gas turbines used for supercharging, may be enumerated as follows:

(*a*) in connection with the Houdry cracking process;
(*b*) in connection with Velox steam boilers;
(*c*) for producing blast air;
(*d*) stand-by and peak-load power plants;
(*e*) locomotives;
(*f*) marine propulsion;
(*g*) stationary power plants;
(*h*) aircraft propulsion.

Experience with applications (*a*) to (*d*) seems to indicate that the gas turbine will gradually assume a more and more important role in all of these four fields.

Locomotives.—In this field the application of gas turbines seems very promising. A number of engine-building concerns, both in this country and in Europe, are conducting development work in this direction. The gas-turbine electric locomotives can easily develop an over-all efficiency higher than the best steam locomotives, since the efficiency of the latter is not very high. At the same time the first cost will be considerably lower than that of diesel locomotives; they will be simpler to operate and require less maintenance than diesel locomotives and possibly even steam locomotives.

Marine Propulsion.—While the gas turbine itself is nonreversible, turbine-electric drives make it entirely suitable for ship propulsion. The further development of the variable-pitch and reversible propeller is another possibility. Finally, since the turbine itself is very simple and short, there is a possibility of developing an installation along the lines of Fig. 32-10*b*. The air compressor will be driven by a separate gas turbine, and the propulsion turbine will be geared to the propeller shaft and consist again of two separate turbines but connected to the same shaft, a forward and an astern turbine, and similar to steam-turbine propulsion. The astern turbine may be built as a one-disk impulse turbine. The light weight, small space occupied, and simplicity of operation make the gas turbine very desirable for ship propulsion.

Stationary Power Plants.—The present design of gas turbines limits their application to units from about 2000 to 10,000 kw. As explained in Sec. 32-7, in connection with Fig. 32-10*b*, such a unit could compete with a steam-turbine plant of equal capacity. Brown Boveri, the oldest in point of experience, is ready to contract for a 10,000-kw unit. Other engine builders are getting ready to build units up to this size. Every little bit of new experience in this field and every little improvement in the over-all efficiency make more probable the wide application in this important field in spite of the present size limitations.

Aircraft.—The bulk and weight of a gas turbine are a handicap to its application for aircraft propulsion. However, present limits of both propeller and gasoline engine seem to be reached at about 2000 bhp. The desire to go beyond this limit because of the tendency toward a further increase in the power developed by an aircraft power plant, without unduly increasing the number of power units used, may result in the use of gas turbines in combination with jet propulsion.

32-9. Problems.—**1.** Using data from Fig. 32-6, find which of the deviations from an isentropic, during compression or expansion, has a greater influence upon the efficiency of a turbine set.

2. Find the ideal thermal efficiency of a combustion cycle with pressure ratios: (*a*) 3.5, (*b*) 4, and (*c*) 5.

3. Using Fig. 32-6 find the thermal efficiency of a turbine set operating without losses between the temperatures of 80 and 1000 F and pressures of 14.7 and 60 psia.

4. Work problem 3 for an admission temperature of 1100 F.

5. Using the same data as in problem 3 find the actual over-all efficiency of the turbine set, assuming compressor and turbine efficiency as 0.84 and 0.87, respectively.

6. Find the temperatures and enthalpies at the points of exit from the compressor and turbine for the conditions of problem 5.

7. Using the same data as in problem 4, find the actual over-all efficiency of the turbine set, assuming compressor and turbine efficiencies as 0.845 and 0.885, respectively.

8. Find the temperatures and enthalpies at the points of exit from the compressor and turbine for the conditions of problem 7.

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Answers to Problems

In general answers are given to all even-numbered problems in each chapter. In a few cases, when an even-numbered problem is only a repetition of a previous problem with different data, the answers are omitted. The author will appreciate it if unavoidable errors are called to his attention.

Chapter 2

2. 382 cu ft/mol.
4. 3.06 psia; 11.64 psia.
6. 49.05 ft-lb/lb-°P; 0.2399 and 0.1769 Btu/lb.-°F.
8. 411F; 108.3 psig.
10. (*a*) 241 F; 0.934 cu ft; (*b*) 115 F; 0.763 cu ft; (*c*) 690 F; 1.525 cu ft.
12. 1.45 Btu.
14. 1.30.
16. 15.8 psig.

Chapter 3

2. 0.222.
4. 36 F.
6. 181.5 Btu/lb-fuel.
8. 188 F.
10. 0.888.
12. 0.195 psia.
14. 236 F; 248 F.
16. 85.6 F.

Chapter 4

2. (*a*) 1 mol C_8H_{18} + 12.5 mols O_2 + 47.05 mols N_2 = 8 mols CO_2 + 9 mols H_2O + 47.05 mols N_2; 1 lb C_8H_{18} + 41.6 cu ft O_2 + 156.4 cu ft N_2 = 27.3 cu ft CO_2 + 1.92 lb H_2O + 160.7 cu ft N_2; (*b*) 1 mol C_8H_{18} + 14.0 mols O_2 + 52.68 mols N_2 = 8 mols CO_2 + 9 mols H_2O + 1.5 mols O_2 + 52.68 mols N_2; 1 lb C_8H_{18} + 222.0 cu ft air = 26.6 cu ft CO_2 + 1.42 lb H_2O + 4.99 cu ft O_2 + 175 cu ft N_2.
4. 197.5 cu. ft.
6. 1097.9 cu. ft.
8. 37.093 cu ft.
10. 754.5 cu ft.
12. 181.8 lb.
14. 238.7 lb.
16. (*a*) 8.51 lb; (*b*) 284.5 cu ft.
18. 17.43; 0.1505.
20. $V_2/V_1 = (4.76n + 1.44n)/(1 + 4.76n + 1.19m)$.
22. (*a*) 1.754; 1.974; 0.137; 10.835; (*b*) 1.975; 0.363; 0.154; 12.208 psia.

24. (*a*) 1.453; 1.362; 0.811; 11.074; (*b*) 1.498; 0.949; 0.836; 11.417 psia.
26. 53.87 ft.-lb/lb-°R; 1.0425.
28. (*a*) 612.54 cu ft; (*b*) 222.48; 198.30 cu ft.
30. (*a*) 577.2 cu ft; (*b*) 291.25; 282.20 cu ft.
32. 3275; 3026 Btu/cu ft.
34. 20,230; 19,040 Btu/lb.
36. 19,900; 18,600 Btu/lb.
38. 19,075 Btu/lb.
40. 0.26%.
42. H_2O = 2.854489 atm; NO = 0.007482; CO = 0.036020; CO_2 = 2.514000; O_2 = 0.009160; H = 0.000163; O = 0.000068; OH = 0.011980; H_2 = 0.009000; N_2 = 14,960; (*b*) 4.81 cu ft.

Chapter 5

2. (*a*) 3560 F; 560 psia; 124.3 psi; (*b*) 4060 F; 896 psia; 150.8 psi; (*c*) 4556 F; 1288 psia; 174.7 psi.
4. (*a*) 1089 psia; 5517 R; 234 psi; (*b*) 1296 psia; 5604 R; 242 psi; (*c*) 1435 psia; 5685 R; 246.4 psi.
6. (*a*) 0.474; (*b*) 0.430.
8. (*a*) 145 psia; 1150 R; (*b*) 700 psia; 5000 R; (*c*) 14.5 psia; 2535 R.
10. (*a*) 145 psia; 1145 R; (*b*) 720 psi; 4970 R; (*c*) 14.5 psia; 2300 R.
12. (*a*) 610 psia; 1620 R; (*b*) 910 psi; 2270 R; (*c*) 4070 R; (*d*) 55 psia; 2270 R.
14. (*a*) 476 psi; 1520 R; (*b*) 3945 R; (*c*) 62 psia; 2600 R; (*d*) 13.1%.
16. (*a*) 455 psi; 1512 R; 3950 R; 78 psi, 2610 R; (*b*) 3.15%.
18. 0.355.
20. 0.48.
22. 0.515.
24. 0.42.
26. 0.274; 0.312; 0.340; 0.350; 0.356.
28. 0.420; 0.453; 0.478.
30. (*a*) b = 1.12: 0.45; 0.49; 0.512; 0.532; 0.548; (*b*) b = 1.40; 0.472; 0.512; 0.534; 0.554; 0.570.

Chapter 7

2. 0.061 in.
4. 13.73.
6. (*a*) 15.1 fps; (*b*) 328 fps; (*c*) 14.1.
8. 14.25; 13.94; 13.58.
10. 13.2; 11.9; 11.1.
12. 7 to 14 sq in.
14. 4.2 to 8.4 sq in.
16. $1\frac{1}{4}$ in.
18. $1\frac{1}{8}$ in.; $1\frac{1}{4}$ in.

Chapter 8

2. (*a*) 581 F; (*b*) 553 F.
4. (*a*) probable; (*b*) certain.

6. 22.1°; 27.2°; 30.5°.
7. (*a*) 0.0047 sec; (*b*) 0.0029 sec; (*c*) 0.0022 sec.
10. 0.3190.
12. 0.3154.
14. 0.3393.

Chapter 9

2. 0.0132 lb; 0.410 cu in.
4. (*a*) 935φ; (*b*) 1015φ; (*c*) 1019φ fps.
6. 1012φ; (*b*) 1100φ fps.
8. 0.0156 lb; 0.469 cu in.
10. 509 fps.
12. (*a*) 0.033 in.; (*b*) 0.015 in.
14. 0.888 in.; 1.776 in.
16. (*a*) 0.358 hp; (*b*) 15.3 hp; (*c*) 0.2%.
18. 7.0 fps.
20. 50.30 lb; 32.41 lb; 1066 psi.
22. 49.27 lb; 42.61 lb; 1615 psi.

Chapter 10

4. 2d.
6. 0.00115 sec; 6.2°.
8. See Fig. 10-6.
10. Sec. 10-4.
12. Sec. 10-5.

Chapter 11

2. 108.0; 117.2; 121.8; 125.5; 123.7; 112.1; 98.5; 65.0; 22.7 fps.
4. 125.0; 137.7; 179.0; 238.0; 312.0; 486.0; 440.0; 279.5 fps.
6. 25.6; 55.6; 70.2; 89.2; 132.6; 140.0; 88.7 fps.
8. 15.7; 20.4; 28.6; 37.9; 39.8; 51.0; 28.5 fps.
10. 3.05; 3.84; 5.80; 8.00; 10.80; 16.1; 22.1; 31.6; 19.3 fps.

12. To solve this problem the pressure difference between the energy cell and main combustion space must be known or assumed; 10 to 20 psi is actually found.

14. Same as in Prob. 12.

Chapter 12

2. 0.913.
4. 0.933.
6. 78 F.
8. 19 F.
10. (*a*) 0.62; 37,400; (*b*) 0.77; 42,000 Btu/sp ft-hr.
12. (*a*) 4.55; 179,600; (*b*) 2.33; 122,300 Btu/hr.
14. 2.0%.

16. 2.85%.
18. 186.7 F.
20. 115 F.
22. (*a*) 8.15; (*b*) 9.38%.
24. 3.37%.
26. 0.824.
28. 0.858; 0.861.
30. 127.6 hp.
32. (*a*) 165.2; (*b*) 166.85 bhp.

Chapter 13

2. 108.46 psig; 716 F.
4. 60.8 psig; 669 F.
6. 1284 F.
8. (*a*) 992 fps; (*b*) 61.3 and 38.7%.

Chapter 14

2. (*a*) 57 psia; 1145 F; (*b*) 542 psia; 1145 F; (*c*) 633 psia; 1175 F; (*d*) 602 psia.
4. (*a*) 662 psia; 1271 F; (*b*) 620 psia.
6. (*a*) 7.60; (*b*) 6.77.
8. 1967 F.

Chapter 15

2. (*a*) 0.061; 0.057 sec; (*b*) 3.85 in.
4. (*a*) Upper cylinder end: exhaust opening 73° or 30%, closing 49°, duration, 0.025 sec; scavenging: opening 49° or 14.2%, duration 0.022 sec; (*b*) lower cylinder end: exhaust opening 60°, closing 40°, duration 0.022 sec; scavenging: opening 40°, duration 0.018 sec.
6. 1.2 in.; 0.002 sec.
8. (*a*) 0.1234 lb or 42%; (*b*) 0.0311 lb/cu ft.
10. 585 F.
12. (*a*) 0.75; 55.4 psi; (*b*) 0.75; 74 psi.
14. (*a*) 0.87; 82 psi; (*b*) 0.905; 84 psi; (*c*) 0.91; 84.5 psi.

Chapter 16

2. 0.915.
4. 0.797.
6. 0.942
8. 0.84; 0.825; 0.78; 0.702; 0.541.
10. (*a*) 0.249; 0.234; 0.207.
12. (*a*) 0.374; 0.369; 0.348; 0.313.
14. (*a*) 0.798; 0.770; (*b*) 0.753; 0.725.
16. (*a*) 0.863; 0.842; 0.827.

Chapter 17

2. 87.3 psi.

4. 1.15-low sp.

6. (*a*) 1028 fpm; (*b*) 5.28; (*c*) med. sp.

8. $9\frac{7}{8}$-in. × $12\frac{1}{4}$-in. × 54 rpm; 16,700 lb.

10. $6\frac{5}{8}$-in. × $6\frac{3}{4}$-in. × 1560 rpm.

12. $12\frac{1}{2}$-in. × $16\frac{1}{2}$-in. × 332 rpm

14. $15\frac{3}{4}$-in. × $21\frac{1}{2}$-in. × 336 rpm.

16. $4\frac{3}{4}$-in. × 6-in. × 1550 rpm.

Chapter 18

2. (*a*) 618 lb/min; (*b*) 795 lb/min.

4. (*a*) 6.667 fpm; (*b*) 10,630 fpm; (*c*) 13.900 fpm; (*d*) 16,800 fpm; (*e*) 19,450 fpm; (*f*) 22,000 fpm.

8. (*a*) int.-14,100 fpm; 22,200 fpm; (*b*) exhaust-27,800 fpm; 43,000 fpm.

10. $d = 3\frac{7}{8}$ in.; $h = \frac{13}{16}$ in.; $t = \frac{5}{8}$ in.; $b = \frac{1}{8}$ in. to $\frac{3}{16}$ in.; $d_0 = \frac{11}{16}$ in.

12. $d = 1\frac{1}{2}$ in.; $t = 0.164$ in.; $b = 0.075$ to 0.100 in.; $d_0 = \frac{5}{16}$ in.

14. (*a*) 2.325 fps; max. 4.65 fps; (*b*) (+)1352 fpsps; (−) 580 fpsps; (*c*) 16 3 lb.

16. (*a*) 13,650 fpm; (*b*) 1.136 psi.

18. $F_{max} = 75$ lb; scale 67 lb/in.; $D = 1.000$ in.; $d = 0.135$ in.; $i = 7.1$ coils.

Chapter 19

2. (*a*) 0.948; (*b*) 9.2 in.; (*c*) 9.75 in.

4. (*a*) suction pressure 47.4% higher; η_v 33.3% higher; η_{ch} 33% higher; η_{id} 13.4% lower; hp 16.7% higher; (*b*) suction pressure 26% lower; η_v about 20% lower; η_{ch} 20% lower; η_{id} 10.3% higher; hp 11.8% lower.

6. (*a*) $h_1 = 35.6$ Btu; 25.2 hp; (*b*) 36.0 hp; (*c*) 48.7 hp.

8. (*a*) 0.891 hp; (*b*) 1.42 hp; (*c*) 2.02 hp.

10. (*a*) 0.28 hp; (*b*) 0.443 hp; (*c*) 0.592 hp.

12. (*a*) 33.5 hp; 48.0 hp; (*b*) 1045 Btu/min; 1670 Btu/min.

14. (*a*) $2\frac{1}{2}$ in.; 2 in.; (*b*) 504 fps.

Chapter 20

2. (*a*) Exhaust opening 62° b.b.c.; exhaust closing 62° a.b.c.; fuel injection 11° b.t.c.; scavenging opening 52° b.b.c.; scavenging closing 52° a.b.c.; (*b*) exhaust opening 70° b.b.c., closing 70° a.b.c.; fuel injection 15° b.t.c.; scavenging opening 55° b.b.c.; scavenging closing 55° a.b.c.

4. Exhaust port opening 67° b.b.c., exhaust valve closing 55° a.b.c., exhaust port closing 67° a.b.c.; scavenging opening 55° b.b.c., scavenging closing 55° a.b.c.; fuel injection 12° b.t.c., both sides, top and bottom same timing in crank angle deg.

6. Exhaust port opening 55° b. crank d.c., exhaust port closing 55° a.d.c.; scavenge port opening 55° b. crank d.c., scavenge port closing 55° a.d.c.; angle $\alpha = 25°$; fuel injection 17° b. in. d.c. of scavenge piston.

8. $\alpha = 15°$; 2.9 in.
10. (*a*) 275 cfm; (*b*) 0.67; (*c*) 0.4.
12. 19,130 fpm.
14. (*a*) 0.448 sq in.; (*b*) 0.4.
16. $a = 24°$; $b = 9°$; $\eta_{mc} = 0.862$; $e = 0.30$.
18. (*a*) 7.3 psig; (*b*) 59%; (*c*) 90%; (*d*) 2560 cfm.

Chapter 21

2. (*a*) 718.4 F; (*b*) 700.9 F.
4. 1.555 sq ft; 12,440 Btu/hr.
6. 1.32 gpm.
8. 161,500 Btu/hr.
10. (*a*) 340 F; (*b*) 293.5 F.
12. 13.2 gpm.
14. 143,300 Btu/hr; 2.95 gpm.
16. 7 ft high, 3.5 ft dia.; $\frac{3}{4}$ in.
18. 2250 cu in.
20. 0.065 in.
22. (*a*) 314.4 F; (*b*) 1038 F.
24. 1034 F.
26. (*a*) 280 F; (*b*) 0.104 in.
28. $s_o = 5200$ psi, compr.; $s_i = 9180$ psi, tens.
30. $s_o = 2457$ psi, tens; $s_i = 2600$ psi, compr.

Chapter 22

2. $\frac{1}{2}$ in.
4. $\frac{9}{16}$ in.
6. (*a*) $\frac{18}{64}$ in.; (*b*) $3\frac{17}{32}$ in.; (*c*) $4\frac{7}{8}$ in.; (*d*) $\frac{11}{64}$ in.
8. $\frac{15}{16}$ in.
11. (*a*) 8; (*b*)$1\frac{7}{8}$.

Chapter 23

2. (*a*) 12 bolts, $1\frac{1}{2}$ in.; (*b*) 12 bolts, $1\frac{3}{8}$ in.
4. (*a*) $\frac{3}{4}$ in. and 1 in.; (*b*) $\frac{3}{8}$ in. and $\frac{5}{8}$ in.; (*c*) rigidity, (*b*)/(*a*) = 1.25.
6. (*a*) $5\frac{7}{8}$ in.; (*b*) $5\frac{1}{4}$ in.; (*c*) $5\frac{1}{8}$ in.
8. 4 studs; 2-in. × $4\frac{1}{2}$ thr./in.

Chapter 24

4. +8.8 psi.
6. (*a*) +135 and −83.5 lb; (*b*) +1214 and −752 lb; (*c*) +1899 and −1175 lb.
8. 15° a.d.c., 13.04 psi; 30°, 19.2; 45°, 18.75; 60°, 18.80; 75°, 17.08; 90°, 15.6 . . .

Chapter 25

2. All balanced.
4. See Table 25-1.

6. All balanced.

8. Four counterweights 448 lb each, c.g. at 11-in. rad.

10. (*a*) Pr. 7: center bearing 503 psi, end bearings 251 psi; (*b*) pr. 8. center bearing (at edge) 21.7 psi; end bearings 125.5 psi.

12. 114.4 vibr/sec.

14. 25.5 vibr/sec; 12th order, 255 rpm.

16. 0.206 in.; 19.35 vibr/sec; 1162 rpm; 2d order.

18. 632 vibr/min.

20. (*a*) 0.40 in.; (*b*) 0.625 in.

22. 38 tn; 20.1 cu yd; 6 or 8 bolts; $d = 1$ in.; $l \geqq 36$ in.

24. $\pm 0.2°$.

26. $d = 1\frac{1}{8}$ in.; 6 or 8 bolts; $l \geqq 42$ in.

Chapter 26

2. 8-40 gpm; pressure pump $d = 2\frac{3}{4}$–$4\frac{3}{8}$ in.; scavenge pump d = 3-$4\frac{5}{8}$ in.

4. 1-$2\frac{1}{2}$ in.; 2-4 in.

6. For 0.02 gpm/hp: $p = 8$; 15 teeth; $D = 1\frac{7}{8}$ in.; face: pressure $\frac{7}{16}$ in., scavenge $\frac{9}{16}$ in.

8. For 0.03 gpm/hp; $p = 8$; $D = 2$ in.; 16 teeth; face: pressure $\frac{7}{8}$ in., scavenge $1\frac{1}{8}$ in.

10. 74.3 rpm.

12. +258.7 psi; −79.2 psi.

Chapter 27

2. 1.125 in.

4. 0.150 in.

6. 3700 psi.

8. Center, 21,000 psi; edge, 20,800 psi.

10. $t_3 = \frac{15}{16}$ in.; $t_3' = \frac{9}{16}$ in.; $t_4 = \frac{1}{4}$-$\frac{5}{16}$ in.; piston rings, $b = \frac{25}{64}$ in., $h = \frac{5}{16}$ in.

12. $t_3 = \frac{45}{64}$ in.; $t_3' = \frac{27}{64}$ in.; $t_4 = \frac{7}{64}$–$\frac{1}{4}$ in.; piston rings, $b = \frac{17}{64}$ in., $h = \frac{3}{16}$ in.

14. $t_3 = \frac{19}{32}$ in.; $t_3' = \frac{3}{8}$ in.; $t_4 = \frac{3}{16}$ in.; piston rings, $b = \frac{13}{64}$ in., $h = \frac{5}{32}$ in.

16. $d = 5\frac{1}{4}$ in.; $d_0 = 2\frac{1}{2}$–$3\frac{1}{2}$ in.

18. $l = 7.5$ in.; $d = 3\frac{1}{2}$ in.

20. $d_1 = 3\frac{1}{4}$ in.; $d_2 = 1\frac{3}{16}$ in.

22. $s_o = 2710$ psi compr.; $s_i = 10{,}150$ psi tension.

24. $s_o = 6160$ psi compr., $s_i = 23{,}650$ psi tension.

26. 4420 psi.

28. (*a*) 1210 psi; (*b*) 865 psi.

30. (*a*) $1\frac{3}{32}$ in.; (*b*) 27,870 Btu/sq ft-hr; (*c*) 6170 psi.

32. 840 psi.

34. small end $\frac{11}{16} \times 1$ in.; $\frac{1}{4}$-in. web; $\frac{1}{4}$-in. ribs; big end $\frac{11}{16} \times 1\frac{3}{16}$ l.

36. $1\frac{1}{8}$ in. with $S_d = 5000$ psi; 1 in. with $S_d = 5500$ psi.

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38. Use 2d section from top in right column, Table 27-2, with $t = 0.5$ in
40. 2 bolts, 1¾ in. with $S_d = 5000$ or 1⅝ in. with $S_d = 6000$ psi.
42. 2 bolts, 1⅞ in.

Chapter 28

2. 1-3-5-7-2-4-6.
4. 1-2-3-4-5-6-7.
6. 5345 psi.
8. $d = 5.0$ in.; $l = 7\frac{1}{2}$ in.; 3d journal 3 in. × 4½ in.
10. $M_1 = 1{,}635{,}000$ lb-in.; $M_3 = 885{,}000$ lb-in.
12. $d_0 = 8\frac{5}{8}$ in.; $d = 12$ in.; $l = 14$ in.; far end $d_2 = 7$ in.; $l_2 = 10$ in.
14. $d = 9\frac{1}{4}$ in.; $l_0 = 8$ in.; web 5 in. × 20 in.

Chapter 29

2. 2600 lb-ft.^{2}
4. 7760 lb-ft.^{2}
6. 961,000 lb-ft.^{2}
8. 389 psi.
10. $D = 20$ in.; $a = 2\frac{1}{4}$ in.; $b = 4\frac{1}{2}$ in.; $W = 151$ lb.
12. 62.4.
14. $p' = 0.05$; $g = 0.025$.
16. $p' = 0.04$; $q = 0.012$.
18. Governor speed 514 rpm; 4 weights 0.55 lb each; $l_1 = 3\frac{3}{4}$ in.
$= 5\frac{5}{8}$ in.; spring 30 lb.

Chapter 30

2. (*a*) 0.575 lb-ft^2; (*b*) 1½ hp; (*c*) 12 cells, 100 amp-hr.
4. (*a*) 600 amp., 1680 watt; (*b*) ¾ hp.
6. 20 cu ft; $h = 8$ ft; $d = 22$ in.
8. 2 tanks 6 ft × 21 in. dia; 1½ or 2-in. pipe.
10. (*a*) 175 psi; (*b*) 3 tanks 15.9 cu ft each; 7 ft × [illegible] dia.
12. 3 tanks, 23.1 cu ft each; 9 ft long × 21.75 in. dia.

Chapter 31

2. 170 bhp.
4. 425 bhp.
6. 525.2 lb-ft; 75.8 psi.
8. 0.786.
10. 0.243.
12. 0.8845.
14. (*a*) 269,000 lb-ft; (*b*) 113,500 lb-ft.

Chapter 32

2. (*a*) 0.300; (*b*) 0.331; (*c*) 0.368.
4. 0.347.
6. (*a*) 214 Btu, 437 F; (*b*) 244 Btu, 560 F.
8. (*a*) 214 Btu, 437 F; (*b*) 256 Btu, 600 F.

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38. Use 2d section from top in right column, Table 27-2, with t = 0.5 in
40. 2 bolts, 1¾ in. with S_d = 5000 or 1⅝ in. with S_d = 6000 psi.
42. 2 bolts, 1⅞ in.

Chapter 28

2. 1-3-5-7-2-4-6.
4. 1-2-3-4-5-6-7.
6. 5345 psi.
8. d = 5.0 in.; l = 7½ in.; 3d journal 3 in. × 4½ in.
10. M_1 = 1,635,000 lb-in.; M_3 = 885,000 lb-in.
12. d_0 = 8⅝ in.; d = 12 in.; l = 14 in.; far end d_2 = 7 in.; l_2 = 10 in.
14. d = 9¼ in.; l_0 = 8 in.; web 5 in. × 20 in.

Chapter 29

2. 2600 lb-ft.^{2}
4. 7760 lb-ft.^{2}
6. 961,000 lb-ft.^{2}
8. 389 psi.
10. D = 20 in.; a = 2¼ in.; b = 4½ in.; W = 151 lb.
12. 62.4.
14. p' = 0.05; g = 0.025.
16. p' = 0.04; q = 0.012.
18. Governor speed 514 rpm; 4 weights 0.55 lb each; l_1 = 3¾ in.; = 5⅝ in.; spring 30 lb.

Chapter 30

2. (a) 0.575 lb-ft^2; (b) 1½ hp; (c) 12 cells, 100 amp-hr.
4. (a) 600 amp., 1680 watt; (b) ¾ hp.
6. 20 cu ft; h = 8 ft; d = 22 in.
8. 2 tanks 6 ft × 21 in. dia; 1½ or 2-in. pipe.
10. (a) 175 psi; (b) 3 tanks 15.9 cu ft each; 7 ft × [illegible] in. dia.
12. 3 tanks, 23.1 cu ft each; 9 ft long × 21 75 in. dia.

Chapter 31

2. 170 bhp.
4. 425 bhp.
6. 525.2 lb-ft; 75.8 psi.
8. 0.786.
10. 0.243.
12. 0.8845.
14. (a) 269,000 lb-ft; (b) 113,500 lb-ft.

Chapter 32

2. (a) 0.300; (b) 0.331; (c) 0.368.
4. 0.347.
6. (a) 214 Btu, 437 F; (b) 244 Btu, 560 F.
8. (a) 214 Btu, 437 F; (b) 256 Btu, 600 F.